AMCP-706-350

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AMC PAMPHLET **AMCP 706-350**

ENGINEERING DESIGN HANDBOOK

WHEELED AMPHIBIANS

HEADQUARTERS, U.S. ARMY MATERIEL COMMAND **JANUARY 1971**

HEADQUARTERS
UNITED STATES ARMY MATERIEL COMMAND
WASHINGTON, D.C. 20315

AMC PAMPHLET
No. 706-350

11 January 1971

ENGINEERING DESIGN HANDBOOK
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LIST OF SYMBOLS

Notes: The same symbol in many instances is used to represent more than one parameter. This is caused by the complex nature of amphibian design requiring many engineering disciplines. The symbols have been chosen to preserve familiar notations found in literature concerning each discipline.

Dimensional units are assigned to each symbol. Dimensionless quantities are indicated by "nd". In some equations, it may be necessary to convert the dimensional units in order to make the formula dimensionally correct; e.g., ft to in. or ft-lb/sec to hp.

A = jet area, ft^2; projected frontal area, ft^2; propeller disc area, ft^2; regular wave amplitude, ft

A_E = expanded bladed area of a propeller, ft^2

A_{comp} = planform area of machinery compartment, ft^2

A_{cool} = planform area of radiator and ducting, ft^2

A_e = effective aspect ratio, nd

A_{eng} = planform area of machinery in machinery compartment, ft^2

A_H = projected frontal area of faired hull, ft^2

A_o = operational availability, nd

A_p = average availability of parts, nd

A_w = projected frontal area of wheels, ft^2

AR = aspect ratio, nd

$ARC(K)$ = Amphibious Research Craft (Hydrokeel)

a = distance or lever arm, ft; acceleration, ft/sec^2

B = maximum molded breadth at the waterline, ft; web material ultimate tensile strength, psi; center of buoyancy; beam, ft

BAR = blade area ratio, i.e., ratio of the actual area of propeller blades to area $\pi D^2/4$, nd

$BARC$ = Barge-Amphibious-Resupply-Cargo (on early designation of LARC LX)

BHP = brake horsepower, hp

BM = vertical distance from center of buoyancy to the metacenter, ft (subscript t denotes transverse metacenter, ℓ denotes longitudinal metacenter)

BOA = overall breadth, ft

BTF = propeller blade thickness fraction ($= t/D$), nd

b = distance or lever arm, ft; span of the foil, ft; undeflected tire width, in.; deceleration, ft/sec^2

b_f = breadth of outstanding flange of a balanced stiffener, in.

CAB = captured air bubble

CL = centerline

C_B = block coefficient, nd

C_D = drag coefficient, nd

C_{D_A} = aerodynamic drag coefficient, nd

C_{D_H} = drag coefficient of faired hull, nd

C_{D_w} = drag coefficient of wheel and wheel wells, nd

C_F = frictional resistance coefficient, nd

C_{f_s} = free surface coefficient, nd

C_{I_T} = transverse inertia coefficient, nd

$C_{L_{o\alpha}}$ = slope of lift coefficient curve for shroud in free stream flow, nd

C_{L_α} = slope of the lift coefficient curve for a shroud enclosing a thrusting propeller, nd

C_R = residuary resistance coefficient, nd

C_T = load coefficient, nd; total resistance coefficient, nd

C_{T_p} = propeller thrust coefficient, nd

$\overline{CI}$ = average cone index of soil, psi

c = coefficient of (soil) cohesion, psi; cord of rudder, ft; shroud length parallel to propeller centerline, ft

D = deep water depth, ft; distance or lever arm, ft; drag force, lb; propeller diameter, ft; undeflected tire diameter, in.

D_{avg} = average vehicle depth, ft

D_1 = slipstream diameter, ft

DHP = delivered horsepower, i.e., the

LIST OF SYMBOLS (Continued)

power actually delivered to the wheels or propeller, hp

$DTMB$ = David Taylor Model Basin

d = depth of stiffener or corrugation; depth of water, ft; distance or lever arm, ft; inside diameter of propeller shroud, ft; tire depth, in.

d_r = rim diameter, in.

E = modulus of elasticity, psi; system energy due to wave action, ft-lb

E_θ = work required to heel vehicle through an angle of θ degrees, ft-lb

EAR = expanded area ratio of a propeller ($= A_E/A$), nd

EHP = effective horsepower, i.e., resistance × speed ÷ 550, hp

F = force, lb; steel temper—as fabricated

F_A = vertical force applied to aft wheels, lb

F_c = stress at failure from axial loading, psi

F_F = vertical force applied at forward wheels, lb

F_N = Froude number, nd

F_u = stress at failure from compressive loading, psi

F_x = force component in a given direction x, lb

F_y = tensile yield stress, psi

f = propeller material constant

f_b = compressive stress from bending load, psi

f_c = compressive stress from axial load, psi

f_d = compressive stress from loads normal to plating, psi

G = average gradient to cone index of sand versus depth, psi/in.; center of gravity

GM = metacentric height, ft

GVW = gross vehicle weight, lb (same as Δ)

GZ = righting arm, ft

g = acceleration constant = 32.2 ft/sec^2

H = draft forward or aft, ft; steel temper—strain hardened; SHP at maximum continuous rating, hp

H_b = height of breaking wave, ft

H_m = mean draft, ft

H_o = deep water wave height, ft

HP_A = power required by auxiliary systems, hp

h = head of water, ft

h_A = atmospheric pressure in feet of water, ft

h_H = depth of immersion of the screw axis, ft

h_v = vapor pressure, ft of water

I = moment of inertia, in.^{4} at ft^4

I_f = moment of inertia with respect to flange, in.^{4}

I_o = mass moment of inertia, ft-lb-sec^2

I_w = moment of inertia with respect to web, in.^{4}

i = moment of inertia of free surface, in.^{4} and ft^4

J = speed coefficient, nd

K = general symbol for constant; baseline (also B_L); rake of propeller blade, in./ft; soil consistency parameter; modulus of deformation

K_T = thrust coefficient, nd

K_Q = torque coefficient, nd

KB = vertical distance from the baseline to the center of buoyancy, ft

KG = vertical distance from the baseline to the center of gravity, ft

KG_v = virtual center of gravity above baseline, ft

k = radius of gyration, ft; ratio of operating hours to total hours, nd

L = length, ft; lift force, lb

L_D = wave length, ft

L_M = length of model ship, ft

L_o = deep water wave length, ft

L_S = length of full size ship, ft

LOA = overall length, ft

$LARC$ = Lighter-Amphibious-Resupply-Cargo

$LCSR$ = Land-Craft-Swimmer-Reconnaissance

$LCVP(T)$ = Landing-Craft-Vehicle-Personnel (test)

LLL = Land Locomotion Laboratory,

LIST OF SYMBOLS (Continued)

U. S. Army-Tank Automotive Command

LVK = Landing Vehicle Hydrokeel type

$LVPT$ = Landing-Vehicle-Personnel Tracked

LVW = Landing Vehicle Wheeled

LWL = length at waterline, ft

LCB = distance of the center of buoyancy forward of the transom, ft; longitudinal center of buoyancy

LCG = distance of center of gravity forward of the transom, ft

ℓ = moment arm of rudder stock, ft; wheelbase, ft

M = metacenter; moment about ship's CG, ft-lb; mass of vehicle, slugs

M_B = bending moment, ft-lb

M_{B_x} = bending moment at x section, ft-lb

$M_{c/4}$ = moment about quarter chord point, ft-lb

Me' = inverse of Eklund mobility factor, nd

M_i = initial turning moment, ft-lb

M_r = moment about rudder stock, ft-lb

MDT = mean downtime, hr

$MTBF$ = mean-time-between-failures, hr

$MT/1$ = moment to trim one inch by the stern, ft-lb

$MTTR$ = mean-time-to-repair, hr

MWR = mean width ratio of propeller blades, nd

m = mass, slug

$\dot{m}$ = mass rate of flow, slug/sec

N = rotational speed, rpm

$NACA$ = National Advisory Committee on Aeronautics

$NDCC$ = nondirectional cross country

NPL = National Physics Laboratory, England

$NUGP$ = nominal unit ground pressure, psi

n = rotational speed, rps; soil sinkage characteristic exponent

O = steel temper—annealed

P = power, hp; grounding force, lb; delivered power, ft-lb/sec

P_o = pressure at a given depth; psi or lb/ft^2; reference pressure pressure, psi; power, hp; grounding force, lb; delivered power, ft-lb/sec

P_m = dynamic pressure, psi

P_r = pitch, at given nd radius r, ft

p = local pressure, lb/ft^2

p_{av} = ATAC average ground pressure, psi

Q = torque, ft-lb; intersection of the grounding force vector with amphibian centerline

QK = height along centerline from Q to the baseline, ft

q = length of shear plane, in.; dynamic head ($=\frac{1}{2}\rho V^2$), lb/ft^2; dynamic pressure of the undisturbed fluid, lb/ft^2

R = propeller radius, ft; radius of turn, ft; reliability, nd; revolutions at maximum continuous rating, rpm

R_{aero} = aerodynamic drag, lb

R^*_{AW} = average added resistance, lb

Re = Reynolds number, nd

R_F = frictional resistance, lb

R_R = residuary resistance, lb

R_T = total resistance, lb

R_W = added resistance in regular waves, lb

R_y = grade resistance, lb

RPM_{max} = revolutions at maximum continuous rating, rpm

r = undeflected tire radius, in.; fraction of total radius divided by R, nd

S = rudder or foil area, ft^2; wetted surface area, ft^2; shear strength, psi

SHP = shaft horsepower, hp

SHP_{max} = shaft horsepower at maximum continuous rating, hp

s = fillet size, in.

s' = slip ratio, nd

T = deep water wave period, sec; natural roll period, sec; propeller thrust, lb; transverse distance or arm, ft; transverse shear strength, psi; steel temper—thermally treated; trim, in.

T_o = turning moment, ft-lb

t = blade thickness, in.; plate or web thickness, in.; thrust deduction, nd; mission time, hr

t_f = flange thickness, in.

LIST OF SYMBOLS (Continued)

t_w = web thickness, in.
U = wind speed, ft/sec; reference velocity, ft/sec
U_{cr} = critical speed, ft/sec
U_{x_1}, U_{x_2} = final and initial velocities of the body in the x direction, ft/sec
u = local velocity, ft/sec
V = forward speed, ft/sec
V_a = propeller speed of advance, ft/sec; initial or upstream velocity, ft/sec
V_{comp} = volume of machinery compartment, ft^3
V_{cool} = volume of radiator and ducting, ft^3
V_d = design speed, mph or ft/sec
V_{eng} = volume of machinery, ft^3
V_j = jet velocity, ft/sec
V_j' = ratio of jet velocity to the advance speed, nd
V_K = vehicle speed, nautical miles per hour (knots)
V_o = initial speed, ft/sec
V_s = ship speed, ft/sec
V_1 = final or downstream velocity, ft/sec
VCI = average rating cone index of field soils for a minimum of 50 passes, psi
v = mean flow velocity over rudder, ft/sec
v_1 = mean slipstream velocity, ft/sec
W = weight of shifted item, lb; average load on one tire, lb; width of blades at one-quarter radius, in.; width of top of corrugation, in.; steel temper—solution heat treated
W_A = hull structure weight, lb
W_B = machinery weight, lb
W_C = drive train weight, lb
W_D = marine propulsor weight, lb
W_E = running gear and suspension weight, lb
W_F = miscellaneous system weight, lb
W_L = load weight—including fuel, crew, and payload, lb
W_M = weight margin, lb
W_T = vehicle weight, lb
$W_{fusion\ line}$ = shear strength at web fusion lines, lb
W_{shear} = web shear strength, lb
$W_{tension}$ = web tensile strength, lb
W_1 = maximum rated load on tire at designed inflation pressure, lb
WCB = West Coast Branch, Amphibian Vehicle Division, Development Center, Marine Corps Development and Education Command
WL = waterline
$W'L'$ = heeled or inclined waterline
w = length between steering pivots, ft; material constant; wake fraction, nd; width of bottom of corrugation, in.
w_i = weight of the i^{th} component, lb
w_x = weight of vehicle in section x, lb
X = distance from base plane to the center of gravity, ft
x = displacement in any direction, ft
x_i = distance from base plane to the center of gravity for the i^{th} component, ft
x = horizontal acceleration, ft/sec^2
y = neutral axis above reference axis, in.
$\ddot{y}$ = vertical acceleration, ft/sec^2
Z = number of propeller blades
z = vertical displacement, ft; sinkage, in.

GREEK LETTERS

α = effective inflow angle, deg; rudder angle of attack, deg
Δ = displacement, lb; gross weight, lb
Δ_v = virtual displacement, lb
$\Delta \cdot GZ$ = righting moment, ft-lb
Δt = duration of impact, sec
∇ = displaced volume, ft^3
δ = rudder helm angle, deg
ϵ = strain, in./in.
η_B = propeller efficiency in nonuniform wake, nd
η_i = ideal efficiency, nd
η_o = open water propeller efficiency, nd
η_p = propulsive efficiency, nd
η_r = relative rotative efficiency, nd
η_T = transmission efficiency—between engine and propulsor, nd

LIST OF SYMBOLS (Concluded)

θ = heel angle, deg

θ_M = maximum roll angle, deg

θ_r = propeller pitch angle at nd radius r, deg

λ = wave length, ft; linear scale ratio, nd

μ = viscosity of fluid, slug/ft-sec

ν = kinematic viscosity ($= \mu/\rho$), ft^2/sec

ρ = mass density, $slug/ft^3$

ρ_s = mass density of water displaced by hull, $slug/ft^3$

ρ_t = mass density of liquid with free surface, $slug/ft^3$

ρg = weight density of fluid, lb/ft^3

σ = cavitation number, nd; normal stress on the soil, psi; tensile or compressive stress, psi

σ_{cr} = critical cavitation number, nd

σ_s = shear stress in weld, psi

σ_y = tensile or compressive yield stress, psi

$\sigma_1, \sigma_2, \sigma_3$ = principal tensile or compressive stress, psi

τ = shear stress, psi

φ = angle of friction, deg; angle of grade with the horizontal, deg

$\Phi^+_{\xi\xi}$ = wave spectral density ordinate, ft^2-sec

ψ_n = nozzle wake fraction

Ω = sweep angle of one-quarter chord line, deg

ω = wave frequency, rad/sec

ω_n = natural frequency of the system, rad/sec

ω_r = angular rudder rate, deg/sec

ω_2, ω_1 = final and initial angular velocities, rad/sec

PREFACE

The Engineering Design Handbook Series of the Army Materiel Command is a coordinated series of handbooks containing basic information and fundamental data useful in the design and development of Army materiel and systems. The handbooks are authoritative reference books of practical information and quantitative facts helpful in the design and development of Army materiel so that it will meet the tactical and the technical needs of the Armed Forces.

This handbook has been prepared as an aid to engineers designing wheeled amphibians. Fundamental design information not readily attainable elsewhere is presented along with requirements and problem areas that are unique to the wheeled amphibian. Part One introduces the principal elements of the wheeled amphibian, design requirements, and a history of amphibian development. The actual design process is described in Part Two in which the amphibian evolves from concept design through preliminary design and finally to detail design. Part Three deals with evaluation and testing of wheeled amphibians as an important link in the design process. Probable developments in future high-speed wheeled amphibian design are described in Part Four.

The material presented in this handbook is composed of data and design information gathered from many sources; however, treatment is limited to one of condensation and summary because of the vast scope of the subject. References appear at the end of each chapter for guidance in acquiring complete information on any topic.

When reference is made to Military Specifications, regulations, or other official directives, it is done so to reveal the existence of these documents. In this respect, the user should obtain the current editions for applications.

This handbook was prepared by Hydronautics, Incorporated, under subcontract to the Engineering Handbook Office of Duke University, prime contractor to the Army Materiel Command for the Engineering Design Handbook Series. Technical supervision and guidance were given by an ad hoc working group under the co-chairmanship of Frank X. Stora and John F. Sargent, Mobility Equipment Research and Development Center, Ft. Belvoir, Virginia.

The Handbooks are readily available to all elements of AMC, including personnel and contractors having a need and/or requirement. The Army Materiel Command policy is to release these Engineering Design Handbooks to other DOD activities and their contractors and to other Government agencies in accordance with current Army Regulation 70-31, dated 9 September 1966. Procedures for acquiring these Handbooks follow:

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Comments and suggestions on this handbook are welcome and should be addressed to Army Research Office-Durham, Box CM, Duke Station, Durham, N. C. 27706.

PART ONE BACKGROUND INFORMATION

CHAPTER 1 INTRODUCTION

1-1 PURPOSE OF THE HANDBOOK

The necessity for an item such as the wheeled amphibian arose from unprecedented demands of WW II to move large amounts of equipment and supplies from ships to storage facilities, or users ashore where no ports existed. Since its inception, the wheeled amphibian has played an important and ever increasing role in the United States Army.

Unfortunately, the evolution of wheeled amphibian design has not been thoroughly documented. There exists no concise, generally accepted explanation of wheeled amphibian design; furthermore, much of the design data and technology in this area is scattered, and in some cases exists only in the memories of the men who were involved in developing these vehicles. This situation could retard the future technological advancement of the wheeled amphibian since much that has been learned may either be forgotten or lost as one generation of designers is replaced by another. In general, without good documentation, cataloging, and information availability; progress is slowed, efforts are duplicated, and money is wasted.

The purpose of this handbook, then, is to present that information from past and current design experience which is most useful for designers of future amphibians.

1-2 DEFINITION

An amphibian is a unique craft designed to be equally well suited for self-propelled operation both in the water and on land. This handbook is concerned only with amphibians which use rubber-tired wheels as the propulsors for land operation.

The fact that an amphibian must inherently possess both land and water capabilities indicates that design compromises must be made since the requirements for a good boat differ from those for a good land vehicle. In most cases, land performance must be sacrificed to achieve seaworthiness and waterborne performance must be sacrificed to achieve reasonable cross-country mobility. Note, however, in Southeast Asia the LARC V was often used extensively in land operations since its cross-country mobility in sand and clay greatly exceeds that of a truck.

The wheeled amphibian is designed to be used exclusively in a logistic-over-the-shore (LOTS) role. Current amphibians have been used on secondary missions such as passenger ferry boat, river patrol boat, and minesweeper. Its primary function is to transfer cargo and, in some cases, personnel from ships anchored off-shore to dispersed locations, or direct to material users, located several miles inland from the beach.

1-3 PRINCIPAL ELEMENTS OF THE WHEELED AMPHIBIAN

The wheeled amphibian is a complex item. A discussion of the principal elements or features follows.

1-3.1 HULL

The hull of a wheeled amphibian is the enclosure which contains the crew compartment, cargo compartment, engine compartment, and numerous other vehicle components. The amphibian hull differs fundamentally from the body of other military automotive assemblies in that it must satisfy certain marine design criteria. Specifically, it must exhibit the required stability and buoyancy characteristics as well as have a configuration that will result in an acceptable speed-power relationship for waterborne operations. In summary, the purpose of the amphibian hull is twofold—it serves as a boat hull and as a truck body.

1-3.2 PROPULSION SYSTEM

The propulsion system is that system of components which converts energy from a fuel source into a form usable by the amphibian and

transmits this energy to those devices which develop the thrust or tractive effort required for propulsion. This system may be considered to include the three major subsystems defined in the paragraphs which follow.

1-3.2.1 Power Plant

The power plant is that part of the propulsion system which converts the energy of the fuel source into mechanical energy. It includes not only the engine but also the fuel, lubrication, cooling, exhaust, hydraulic, and electrical systems. The engine types most commonly used are internal combustion types, i.e., gasoline, Diesel, and gas turbine. A detailed discussion of the various types of power plants appears in Chapter 7.

1-3.2.2 Power Train

The power train is that system of components which transmits the useful energy produced at the output shaft of the power plant to the ultimate point of application, i.e., the propulsors. The wheeled amphibian normally has two sets of propulsors—wheels for land operation, and a marine propulsive device, usually a screw propeller, for waterborne operation. This means that the wheeled amphibian has two power paths. In most cases, the power path is common for either mode of operation up to a transfer case. At this point, there are separate drive shaft take-offs for the wheels and the propeller. The major components of the power train are clutches, transmissions, transfer cases, drive shafts, differentials, axles, and brakes. On the other hand, power transmission may possibly be effected by electrical, hydraulic, or pneumatic systems with quite different components.

1-3.2.3 Propulsors

The propulsors are those devices to which the power train delivers the mechanical power developed by the power plant. For overland operations, the propulsors are the wheels which apply the required tractive effort to the ground. The waterborne propulsor is usually a device which develops thrust by acceleration of water in the direction opposite to the desired direction of vehicle motion. The most successful types are the waterjet, the paddle wheel, and the screw propeller. Variations of these three basic types have been theoretically or experimentally considered. All standardized U. S. wheeled amphibians to date have used screw propellers.

1-3.3 SUSPENSION SYSTEM

The suspension system is that series of mechanical components between the vehicle frame or hull and the ground. The function of this system is to provide a flexible support for the assembly, which has good stability and control characteristics and which will allow the required overland speed to be attained. Though there are many types of suspension systems, the one that will receive the most attention in this handbook will be the "rigid" type in which the wheel assemblies are the only element in the system.

The suspension system is functionally useful only in the land mode. However, it must be considered in the waterborne mode since the wheels and the wheel wells are a major source of resistance and greatly influence the waterborne performance capability of the vehicle.

1-3.4 STEERING SYSTEM

The steering system is that assembly of components and linkages which enables the operator to control the direction of the vehicle. For land operation, steering systems are normally designed to provide for two-wheel steering or selective two-wheel, four-wheel, and oblique steering. For waterborne operations, steering can be accomplished by rudders or trainable thrust devices, by training the wheels (as for land operation), or by combinations of these methods.

1-3.5 AUXILIARY SYSTEMS

Auxiliary systems are those systems which cannot be included in any of the previous major categories because they are not related to any one of these categories, or because they are support systems related to several of them. Of those auxiliary systems discussed in Chapter 7, the electrical and hydraulic systems are the most important.

1-3.5.1 Electrical System

The electrical system of an amphibian performs the same type of functions and services as the electrical system of any military automotive vehicle. In general, electrical power must be supplied to communications equipment, interior and exterior lighting systems, engine starting and ignition systems, heaters, pumps, fans, etc. While the standard Army automotive vehicle electrical system is a 24-V DC single-wire system incorporating a generating, distribution, and storage system; the more costly two-wire marine wiring system is desired for amphibians due to its waterborne operating environment.

1-3.5.2 Hydraulic System

A hydraulic power system is commonly used on amphibious vehicles to perform a host of auxiliary powering functions. The most common uses of hydraulic systems are to provide power to the steering system, the cargo winch, the ramp cylinders, the cargo and bilge pumps, and the cooling fan motors.

1-3.6 PAYLOAD

The wheeled amphibian is primarily a logistic vehicle; its payload is cargo. This cargo may be small vehicles, personnel, or containerized, palletized, and loose general cargo. The primary aim of logistic amphibian design is to produce a vehicle which can effectively and efficiently deliver a required amount of cargo.

1-3.7 FUEL

The required fuel load (subject to operating range) is of the same order of importance as the payload carrying ability. Usually, for a given amphibian, the total weight of cargo and fuel that can be carried is a constant. Thus, the more fuel that is required to perform the mission(s), the less will be the weight of cargo which can be carried. In some cases, the fuel consumption characteristics of a power plant will be the primary consideration for its selection or rejection.

CHAPTER 2 FACTORS AFFECTING THE DESIGN OF WHEELED AMPHIBIANS

SECTION I MILITARY REQUIREMENTS

2-1 THE MILITARY MISSION AND ENVIRONMENT

The use of amphibians for logistic support was initiated during WW II due to the lack of suitable port facilities and because unfriendly areas were not directly accessible to large ocean-going vessels. The LARC V, the LARC XV, and the LARC LX are the newest post-war, wheeled, logistic amphibians and they follow the initial concept of the DUKW which was begun in 1942.

The development of true amphibians is motivated by the advances in weaponry which have prohibited the establishment of large beachheads. Thus, the present concept calls for the amphibians to move up and over the beach to dispersed materiel dumps located a moderate distance inshore from the beach. Amphibians are used to transport materiel from ships located up to 10 miles off-shore to destinations as far as 5 miles inland. General wartime planning figures indicate that approximately 80 percent of all cargo entering a theater of operations will be delivered through dispersed beach terminals, and 20 percent will move through established terminal facilities.

To satisfy dispersion requirements, the average beach terminal will work only two ships at a time. As a general rule, these ships will be anchored several miles off-shore, and amphibians will transport cargo over the beach to multiple transfer points or directly to users. These locations will be as close to the beach as practicable, while avoiding the establishment of congested materiel depots which are lucrative targets for the enemy.

Each of the two-ship discharge sites will support a division tonnage requirement of 1,440 short tons per day. Normally, five of the discharge sites will be dispersed a minimum of 5 miles apart, under terminal command control, to support a force of five divisions (7,200 short tons per day). These sites will be operated by two terminal service companies (each having a 720 short tons per day receiving rate), one light amphibian company (1,080 short tons per day delivery rate), and one heavy amphibian platoon (360 short tons per day delivery rate), all under terminal battalion control. Other small craft (barges, picket boats, etc.) will be assigned as required.

The lighterage assignments given are based on an average situation in which approximately 25 percent of all cargo moving over the beach will be vehicular or heavy equipment. Containerized, palletized, and loose general cargo comprise the remaining 75 percent. The tonnage capabilities of the amphibian units are specifically tailored to this 75-25 ratio of the 1,440-ton division requirement. The heavy amphibian delivers the outsized items (comprising the 25 percent) and the light company handles the remainder (comprising the 75 percent).

Thus, any improvement in the characteristics of the amphibian which results in increased speed, mobility, or payload will provide quicker, more efficient supply to the forces ashore.

This handbook is intended for use by designers who are relatively inexperienced in the design of wheeled amphibians. Therefore, it is of the utmost importance that the severity of the military environment be appreciated. Military vehicles must be inherently more rugged and reliable than commercial equipment.

Commercial vehicles are not designed to meet the rigors and widely divergent demands of the military environment. In fact, the development of the commercial automotive vehicle is attributable, to a great extent, to safe smooth roads with good traction. Such roads are virtually nonexistent in many combat zones.

The military vehicle must be capable of operating with satisfactory mobility over terrain which ranges from deep mud, sand, and steep rocky topography to paved highways. It must be reliable in arctic climates as well as in tropical regions.

A wheeled amphibian must not only be reliable under the above stated conditions. It

must also be seaworthy in Sea State 3, survive the loads imposed by plunging surf, and operate in a highly corrosive environment when waterborne. Since it is a military vehicle, it may be called upon to effect landings under enemy fire.

Perhaps the most important characteristic of the military environment is the lack of highly skilled operators for wheeled amphibians. In most cases, the military operator is young and inexperienced; and in combat, it is only natural to worry more about one's safety than the proper operation of complicated and sophisticated controls. Other characteristics of the military environment are excessive dirt, subjection to explosive impact, and perhaps long periods of operation with little or no maintenance.

The major features of the military environment are listed:

a. High shock and vibration produced by off-road operations over rough terrain, air drop operations, high explosive blast, and ballistic impact

b. Extreme temperature ranges, extending from arctic to tropical

c. Operations under conditions of extreme dust

d. Operations in deep mud

e. Operations in ice and snow

f. Amphibious operations through heavy seas and plunging surf

g. Operations under conditions conducive to corrosion and fungus growth. Corrosion is a particular problem for the wheeled amphibian operating in the marine environment.

h. Operations involving long steep grades and side slopes

i. Extended operations under conditions of low speed and high load

j. Operator abuse in the form of overload, misuse, improper maintenance and neglect

A more detailed description of the military environment may be obtained from Refs. 1 and 16.

2-2 ESTABLISHMENT OF MILITARY REQUIREMENTS

2-2.1 GENERAL

The generation of military requirements for Army materiel in the form of Qualitative Materiel Development Objectives (QMDO), Qualitative Materiel Requirements (QMR), or Small Development Requirements (SDR), is provided by the U. S. Army Combat Developments Command (CDC).

These QMDO's, QMR's, and SDR's are established as a result of needs created by combat development studies for the determination of Army doctrine (how the Army of the future is to be organized and equipped, and how it is to fight).

New requirements documents are prepared by CDC and also in draft form by any agency or command of the Army. A QMDO may be initiated at any stage in the research and development cycle and may result from a combat development study, operational experience, development experience, technological breakthroughs, etc. QMR's and SDR's may be processed directly if the development is known to be feasible. However, if feasibility is uncertain, the usual procedure is to prepare a QMDO to provide the necessary data. Then a QMR, with its associated military characteristics, is formulated.

Detailed procedural information regarding the life cycle of Army materiel is provided in Ref. 2.

2-2-2 QUALITATIVE MATERIEL DEVELOPMENT OBJECTIVES (QMDO)

A QMDO is a Department of the Army (DA) approved statement of military need for the development of new materiel, the feasibility of which cannot be determined sufficiently to permit the establishment of a qualitative military requirement. The QMDO normally originates from requirements generated by combat development studies conducted by CDC. The QMDO as previously defined, pertains to items of materiel which are not sufficiently within the state-of-the-art to permit immediate development and, therefore, require further research. In general, these items are three to eight years away from the firm establishment of their military characteristics. The QMDO states in general terms the objective, the operational and organizational concepts for employment, and a description and full justification for the item.

The Army Materiel Command (AMC) Director of Research, Development and Engineering is responsible for processing all new materiel

objective documents and for insuring that adequate QMDO plans are prepared. After the draft QMDO is reviewed by other interested agencies, it is revised and submitted by CDC to DA for approval and staffing. Upon DA approval, the QMDO is published in the Combat Development Objectives Guide (CDOG) which contains all approved DA operational objectives, organizational objectives, QMDO's, QMR's and SDR's. Within the DA staff, the Chief of Research and Development will assign responsibility to a development agency whose purpose will be to prepare a QMDO plan which will insure that appropriate research and development is conducted in support of the QMDO. Upon approval of the QMDO plan by the DA Chief of Research and Development, the AMC will assign prime, attendant, or sole responsibility for the QMDO to one or more of its commands or laboratories to perform the necessary work.

2-2.3 QUALITATIVE MATERIEL REQUIREMENTS (QMR)

After research and exploratory development have progressed to the point where the AMC feels it is possible to translate the knowledge and data acquired by the laboratories—in support of the QMDO—into a feasible system, it will recommend that the CDC prepare and issue a QMR. The QMR is directed toward the attainment of new or substantially improved materiel which will significantly advance the Army's ability to accomplish its mission.

Unlike the QMDO, the QMR is much more specific in describing the requirement. It states major materiel needs in terms of military characteristics and priorities, and relates the materiel to the operational and organizational context in which it will be used. The QMR is prepared as soon as possible after the need is established and probable feasibility is determined. The CDC staffs the QMR within its own organization for organizational, operational, and doctrinal concepts; with the AMC it assures that the proposed requirements are within the state-of-the-art. It also checks with the Chief of Personnel Operations for personnel implications, and checks with the United States Continental Army Command for training and training device implications. The QMR is then forwarded by the CDC to the DA for approval. Concurrently, the AMC would normally prepare and forward a Technical Development Plan (TDP) to the Office of the Chief of Research and Development (OCRD) defining the approach for achieving the QMR. The TDP should contain a history of the project, a detailed plan for its development, a test program, maintenance and reliability concepts and goals, firm scheduling and funding plan, a quality assurance plan, a configuration management plan, a new equipment training plan, and perhaps a procurement plan.

The new QMR will not be approved in the DA unless the TDP indicates that the following prerequisites have been fulfilled:

a. Primarily engineering rather than experimental effort is required.

b. "Building block" components and technology are sufficiently in hand.

c. A thorough trade-off analysis has been made.

d. The best technical approach has been selected.

e. The mission and performance envelopes are defined and optimized for technical feasibility and cost effectiveness.

f. The cost effectiveness of the proposed item has been determined to be favorable in relation to the cost effectiveness of competing items (on a DOD-wide basis).

g. Cost and schedule estimates are credible and acceptable for full scale development.

Upon DA approval, the QMR will be published in the CDOG.

2-2.4 SMALL DEVELOPMENT REQUIREMENTS (SDR)

An SDR states an Army need for the development of equipment which can be developed in a relatively short time, and does not warrant the major effort required in developing a QMR. An SDR is normally considered to be an appropriate requuirements document if:

a. The development item is of proven feasibility,

b. Time required for development is 2 years or less, and

c. Research, development, test, and evaluation costs will not exceed $2.5 million, and the projected investment of "Procurement

of Equipment and Missiles, Army" funds (PEMA funds) will not exceed $10 million.

SDR's are staffed and approved by Headquarters, DA, in a manner similar to that prescribed for QMR's. Recommendations for establishing SDR's contain a brief description, purpose, priority, justification of need, and cost of development. Projects initiated as a result of SDR's are handled in the same manner as they are for QMR's.

SECTION II ENGINEERING REQUIREMENTS

2-3 THE MISSION OF ARMY WHEELED AMPHIBIANS

Amphibians are a unique class of vehicles capable of extended operations over both land and sea. Unfortunately, some of the criteria for good water performance conflict with criteria for good land performance. As a result, any amphibian design must be a compromise, a hybrid vehicle which is not likely to exhibit either optimum land or water performance. Hence, for efficient utilization, amphibious vehicles should be used where their special characteristics are clearly indicated, i.e., for missions which require both land and water operation.

The primary functions of Army amphibians may be divided into two categories—(1) tactical, and (2) logistical support. Tactical uses include direct support of assault and combat operations, including transport of personnel, supplies, light artillery, ammunition, missiles, etc., across beachheads and rivers. The seaworthiness and surfing ability of amphibians can be used to achieve strategic and tactical advantage by operating at places and in weather considered impossible for landings by conventional landing craft. Logistic uses involve supply and resupply of general cargo, from deep water shipping anchored offshore, directly across beaches to inland transfer points. Amphibians are not limited to these functions. Their versatility renders them capable of many diverse operations, as determined by circumstances and the operator's imagination.

Tracked amphibians are generally less vulnerable to enemy fire than wheeled amphibians. On the other hand, wheeled amphibians are capable of higher speeds both in water and over hard, smooth soil and roads. Hence, tracked amphibians are generally better suited to tactical support, whereas wheeled amphibians are preferred for logistical support. Since this handbook is directed to the design of wheeled amphibians, greater emphasis is placed on requirements for cargo transport.

2-4 OPERATIONAL REQUIREMENTS AND LIMITATIONS

2-4.1 GENERAL

The operational requirements for an amphibian define its intended mission or use. They range from basic requirements—such as payload, capacity, and roll stability—to special requirements such as cold weather starting and air drop capabilities. Proper selection of the requirements demands careful consideration of the limitations involved. The requirements must never be specified arbitrarily since this procedure leads to mutually prohibitive requirements that preclude the development of a satisfactory vehicle. For example, if payload is specified, the gross vehicle weight should not be arbitrarily stipulated.

Amphibians are subject to limitations imposed by nature and by the state of technology. The laws of nature cannot be opposed. However, natural laws which impose restrictions on present amphibians may possibly be circumvented in future designs if the problem is attacked with initiative and imagination. Any such development would, of course, be classified as a major breakthrough. There has been no such breakthrough in amphibian design since the development of the first WW II DUKW. Limitations imposed by the state of technology will continue to be surmounted with future development and technological progress. For example, advances in engine design will result in lighter, more efficient power plants so that future amphibians may have greater range and speed capabilities with no increase in weight.

Operational requirements and limitations for amphibians will continually change with changes in mission and use, and advances in vehicle technology. The designer should be aware of the process and underlying principles used to determine the amphibian requirements. In many instances the designer must initiate changes in the requirements in order to alleviate conflicts within the requirements or to bring the requirements in line with the limitations imposed by the current state of technology. The requirements should be written to allow for reasonable, low risk advances in technology so that the designer is induced to incorporate these advances in developing a new amphibian. In this manner, progress is assured in the evolution of amphibian vehicles. On the other hand, unreasonable requirements will result in wasted effort and in some cases the requirement may be met, but the added capability does not justify the necessary expense.

The paragraphs which follow are presented in order to identify some of the basic areas in which requirements are specified, the nature of requirements, and the underlying principles and philosophy involved in their determination.

2-4.2 WATERBORNE PERFORMANCE

A typical mission for a wheeled amphibian is the transfer of supplies from a ship stationed offshore to an inland depot. Waterborne performance requirements are specified so that the proposed amphibian will be able to receive cargo at the ship's side and carry it to the beach quickly, efficiently, and safely.

2-4.2.1 Speed and Power

Minimum speed and power requirements for a logistic amphibian may be based upon considerations for safe operation through surf. Speed and power are, of course, closely related. It is desirable to have adequate speed so that the amphibian can match the speed of waves advancing toward the beach. It must have sufficient power to pass through these same waves on the return trip. Particular minimum values are not generally specified, but are left to the discretion of the designer. It is more likely that design speed will be selected on a basis of optimization studies of the entire logistic problem rather than on a safety requirement basis.

The type of power plant may be either specified or specifically limited for such reasons as maintenance and safety. For example, gas turbines may not be allowed because personnel in the field are not experienced in the special maintenance procedures for these engines. Gasoline engines may be prohibited because of the fire hazard.

A minimum allowable speed may be specified for amphibians that will perform tactical missions. The reasons for high speed in this case are obvious.

A requirement which is often specified is a minimum range or endurance for continuous operation. The speed and power of the design are indirectly affected by this type of requirement and others which apply to the various aspects of performance.

2-4.2.2 Maneuverability

Maneuverability requirements are likely to be in qualitative terms, viz., able to maneuver alongside a ship in Sea State 3, safely negotiate 10-ft plunging surf, etc. These requirements imply rather than specify a degree of maneuverability. This approach is understandable in view of the difficulty in applying meaningful quantitative measures to such a broad concept as maneuverability.

Specific maneuvering characteristics may also be stipulated. Minimum bollard pull, maximum allowable turning diameter (tactical diameter), and maximum allowable heel during a turn are common specifications. The difficulty with these characteristics is that they are only qualitatively associated with good maneuverability. Furthermore, safety and the ability to pass through surf probably depend as much on operator skill as on vehicle characteristics.

2-4.2.3 Seakeeping

Seakeeping requirements, like those for maneuverability, are apt to be qualitative in nature. The same requirement may be applicable to maneuverability and seakeeping, i.e., the ability to negotiate a 10-ft plunging surf implies a certain degree of seakeeping ability as well as maneuverability.

Other requirements which relate to seakeeping are more specific. Reserve buoyancy, statical stability, and range of stability are all good indicators of relative seakeeping ability and minimum criteria may be specified, based on experience with successful amphibious operations and general practice. Self-bailing cargo decks or automatic bailing of cargo wells and watertight hatches are additional requirements that enhance seakeeping ability.

2-4.3 OVERLAND MOBILITY AND PERFORMANCE

Overland performance requirements are prescribed to assure that the amphibian will be able to negotiate any reasonable terrain between the water and the inland depot. Increased off-road mobility is generally the strongest motivation for these requirements, and safety is not as demanding a consideration here as in the case of waterborne performances.

2-4.3.1 Soft-soil Mobility

A requirement that simply states that a vehicle must be able to negotiate mud and sand or some other soft soil is too general to be of practical use to the designer because there are so many different types and grades of soil. A better requirement would specify one or more types of soil and a particular minimum index for each. A soil index is a quantitative measure of some physical property of the soil which can be empirically related to the vehicle trafficability. Several soil indexes are tabulated in Table 7-44. It should be noted that the indexes do not take into account all of the soil properties which affect mobility and are, therefore, only approximate guides.

Minimum soft soil mobility requirements that are given in terms of mobility (soil) indexes should be based on experience with operational amphibians and the proposed mission of the new design, i.e., the types and grades of soil most likely to be encountered.

2-4.3.2 Rough Ground and Obstacle Performance

Rough ground performance requirements are usually specified in terms of a minimum acceptable speed over some standard, reproducible, hard ground profile. Obstacle performance is specified by the ability to pass over a ditch, bump, or vertical wall of particular size and geometry. Speed is not of particular importance here. These ground profiles and obstacles are not necessarily representative of the conditions that an amphibian will face in the field. However, such requirements are easily confirmed by proof tests, and performance capability over these simple courses is indicative of the relative mobility of the amphibian over rough terrain.

2-4.3.3 Slope Performance

Slope performance requirements for military vehicles have traditionally been specified as the ability to negotiate a 60-percent grade and 30-percent side slope of smooth dry concrete. (The tangent of the angle between the ground and the horizontal plane is expressed as a percentage. Thus, a 100-percent grade corresponds to a 45 deg angle.) These performance requirements, like those for obstacles, do not represent conditions that the amphibian will meet in the field, but they do set a standard which is easy to confirm by prototype testing.

Side slope requirements are used mainly to specify the degree of stability that a vehicle must possess. Grade requirements present a good test of overall vehicle toughness. The specifications are usually written to include the capability of the brakes to hold the vehicle on the grade, and the ability to climb and descend the grade in both forward and reverse gear. Grade tests put large loads and stresses on the power train, the engine and its cooling system, and the suspension system.

2-4.3.4 Maneuverability

A minimum turning diameter is generally specified as the requirement on maneuverability. Other requirements which may be made include the maximum safe speed at which a specified turn may be negotiated, four-wheel steering, and crab steering.

2-4.3.5 Dimensional Requirements

Limitations on overall vehicle length, width, and height are generally based on transportability criteria and are discussed in par. 2-5.

Brake angle, angles of departure and approach, and bottom clearance may be specified. However, these characteristics are mainly associated with obstacle clearance and may be superfluous if the requirements for obstacle performance are comprehensive.

2-4.3.6 Improved Road Performance

The requirement for improved road performance is simply a specification for maximum sustained speed. This requirement was compromised for recent amphibian designs because it conflicted with the requirements for waterborne performance which were considered more important. Rigid suspension system design rather than power was the limiting factor for these vehicles. A reasonable speed for overland travel in convoys is 30 to 35 mph.

2-4.4 RIVER BANK PERFORMANCE

To this date, there probably have been no requirements written specifically for river bank performance of amphibians. To some degree, requirements for soft soil mobility and obstacle performance provide for this special capability. This is an area that is in need of research and development. It is likely that more comprehensive riverine requirements may be developed for future amphibian designs.

2-4.5 PERFORMANCE WHEN ENTERING AND LEAVING LANDING (LST) AND DOCK-TYPE (LSD) SHIPS

Capability for entering and leaving particular classes of ships is often specified. Such a requirement should be based on the intended use of the amphibian and, particularly, its proposed method of deployment.

2-4.6 CARGO-CARRYING AND HANDLING REQUIREMENTS

The flexibility and usefulness of a logistic amphibian are greatly enhanced if it is capable of self-loading and unloading of cargo. Thus, requirements may be written to this end prescribing that loading ramps and/or winches be incorporated in the design.

2-5 TRANSPORTABILITY REQUIREMENTS AND LIMITATIONS

2-5.1 GENERAL

The design of wheeled amphibians is governed by various restrictions which limit and control their size, weight, and shape. Among these restrictions are those imposed by the need for unrestricted transportability in any or all of the transportation modes—rail, air, road, and ship. The transportability requirements are usually specified in the military characteristics for each developmental vehicle.

The policy of the Department of Defense with regard to transportability of items of materiel, as set forth in AR 705-8, Ref. 3, states that transportability must be a major consideration when establishing the priority of design characteristics for a new item of materiel or equipment, to readily permit transport by available means. Special or unique arrangements of schedules, right-of-way, clearances, or other operating conditions, will be undertaken only in exceptional cases and after first obtaining approval from the appropriate transportation agency, including the following designated by the Department of Defense:

a. Dept. of the Army: DCSLOG, Director of Transportation, U. S. Army, Washington, D. C. 20315

b. Dept. of the Navy: Naval Supply Systems Command Headquarters, Washington, D. C.

c. Dept. of the Air Force: Headquarters, Air Research and Development Command, Washington, D. C.

d. U. S. Marine Corps: Commandant, Marine Corps, Washington, D. C.

The designer must consider the limitations of the particular modes of transportation specified or selected for the vehicle to determine the maximum allowable dimensions and weight for an amphibian. The sources which are at the disposal of the designer to determine these limitations are:

a. The Highway Weight and Size Limitations established by the Federal Aid Highway Act of 1956.

b. The highway regulations of the individual States of the United States.

c. The physical limitations of highways in foreign countries.

d. The Outline Diagram of Approved Limited Clearance of the Association of American Railroads

e. Weight limitations of individual carriers, available in the current issue of the Railway Line Clearance publication for individual railroads of the United States, Canada, Mexico, and other nations

f. Diagram of the Bern International Rail Interchange Agreement with weight limitations applicable to the railroads of individual countries for all items that may require transportation by rail in foreign countries

g. Loading and storage limitations of ocean vessels, and Army and Navy cargo handling procedures which are applicable

h. Regulations of the Department of Transportation (U. S. Coast Guard), the U. S. Army Mobility Equipment Command, and the U. S. Navy, covering water transportation

For convenience some of the main restrictions are presented in the paragraphs which follow. For more complete information refer to Refs. 1 through 9.

2-5.2 UNRESTRICTED HIGHWAY TRANSPORTABILITY

Ref. 1 prescribes maximum dimensions and weights for pneumatic-tired, highway, and off-the-road type vehicles for unrestricted highway operations. Oversize and/or overweight vehicles, as defined by the state roads commission of a particular state through which the vehicle is to be moved, are subject to certain procedures which usually involve the procurement of a permit with payment of a fee and with movement restricted to certain highways at specific times of the day.

2-5.2.1 Width

The maximum overall width of wheeled vehicles for unrestricted highway movement is 96 in.

2-5.2.2 Height

The maximum overall height of wheeled vehicles designed for unrestricted highway operation in the continental United States is 12.5 ft. The maximum allowable overall height for overseas highway operations is 11 ft.

2-5.2.3 Length

The maximum overall length of a wheeled vehicle comprised of a single, nonarticulated unit (in most cases wheeled amphibians will fall into this category) shall not exceed 35 ft. For wheeled vehicles comprised of an articulated double unit, such as a tractor-trailer, the overall length should not exceed 50 ft.

2-5.2.4 Axle Loading

An axle load is defined as the total load transmitted to the road by all wheels whose centers are included between two parallel transverse vertical planes 40 in. apart, extending across the full width of the vehicle. For the continental United States and overseas, the maximum axle load (subject to gross weight limitations) is 16,000 lb for axles spaced between 3½ ft and 7½ ft. For vehicles, designed for highway operation in the continental United States, and with axles spaced more than 7½ ft from the nearest adjacent axle, the maximum permissible load is 18,000 lb; for vehicles designed for overseas operations, it is 16,000 lb.

2-5.2.5 Gross Weight

The maximum permissible gross weight for a wheeled vehicle designed for highway operations (subject to axle load limitations) is 36,000 lb for vehicles having a distance of 10 ft or less between the extreme front and rear axles, and increases 850 lb for each foot of axle spacing over 10 ft to a maximum of 60,000 lb. This maximum gross weight exceeds the statutory limits of some states in the U.S.A. and special permits are required.

2-5.3 RAIL TRANSPORTABILITY

The requirements for transportability by railroad are set by certain clearances determined by bridges, tunnels, platforms, power transmission poles, and various wayside structures that may be encountered. Required clearances have been compiled and used to produce the clearance outline diagrams in Ref. 1. These diagrams set the maximum allowable cross section of a vehicle as loaded upon a railway car for shipping. In cases where vehicles of larger cross section are to be transported,

they must be reducible in size by suitable disassembly to meet the diagram limits.

2-5.4 AIR TRANSPORTABILITY

When designing military vehicles for airlift, consideration must be given to the size, weight, and location of the vehicle's center of gravity; the size, location and configuration of the aircraft loading apertures; the size and configuration of cargo compartments (including limiting features that may prevent the full use of available space); the strength of the aircraft floor and loading ramp; the location, capacity, and type of tie-down fittings used; the allowances for loading controls and auxiliary equipment; and the air transportability requirements specified in MIL-A-8421A (USAF), Ref. 4. Each vehicle should be designed to be transportable in a maximum number of aircraft types to permit maximum use of available aircraft and to alleviate the possible shortage of a particular type. When designing vehicles for aerial delivery (airdrop), vertical and horizontal clearances in aircraft must be considered; in addition, the designer must make allowances for the weight and space taken up by the aerial delivery equipment and for the motions of the "package" as it leaves the aircraft. Additional data and guidance can be found in Refs. 5, 6, 14, and 15.

2-6 CLIMATIC ENVIRONMENT CRITERIA

Amphibians are frequently subjected to harsh corrosion environment resulting from frequent operation in and out of salt water. This and other environmental requirements are discussed in Ref. 7.

2-7 ELECTRICAL SYSTEM SPECIAL REGULATIONS

The Department of the Army has established a 24-V DC system as a design standard, as specified in AR 705-19, Ref. 8.

General design considerations—such as generating systems, voltage control, wiring harness, weatherproofing, suppression of radio interference, etc.—are discussed briefly in Ref. 1 and detailed treatment of some of the above factors may be found in Ref. 9.

2-8 SPECIAL REGULATIONS FOR FUELS, LUBRICANTS, AND HYDRAULIC FLUIDS

Specifications for fuels, lubricants, and hydraulic fluids are given in par. 3-2.5 of Ref. 1, and Ref. 13.

2-9 MAINTENANCE REQUIREMENTS

2-9.1 GENERAL CONSIDERATIONS

The design of a wheeled amphibian must provide an available and reliable tool for the Army. These two objectives are a function of the design as well as the maintenance performed after the vehicle is built.

Reliability can be designed into the vehicle and thus reduce maintenance. Maintenance that is relatively easy to perform because of accessibility or other criteria discussed in the paragraphs which follow will make the vehicle available and more reliable. Thus availability, reliability, and maintainability are all interrelated.

2-9.2 MAINTAINABILITY CRITERIA

Maintainability is a characteristic of good design. A vehicle is maintainable if the specified maintenance can be carried out routinely so that the vehicle continues to meet stipulated performance requirements. The definition of maintainability must be related to some measurable quantities which will significantly influence the designer's layout or machinery arrangement, or choice of components. These quantities are realistic descriptors: mean-time-to-repair (MTTR); mean-time-between-maintenance (MTBM); and mean downtime (MDT). This type of approach to maintainability is described in Ref. 10.

To design maintainability into a wheeled amphibian, the designer should hypothesize possible failures and determine procedures for repair. This process will immediately present the major factors involved, including:

a. Possible cause of failure
b. Access
c. Tools
d. Repair parts
e. Fasteners

f. Specialists required for a complex repair.

Some of the specific criteria which are directly applicable to the maintainability of wheeled amphibians are given in par. 3-3.1 of Ref. 1.

2-9.3 RELIABILITY CRITERIA

Wheeled amphibians will tend to become more complex and sophisticated. The more complex a system becomes, the greater the tendency to failures. A complex system will not necessarily be an unreliable one, but complexity does imply that the designer's work must reflect a greater than usual concern for component reliability.

Reliability is defined as the probability that materiel or equipment will perform its designated function for a prescribed period of time under stated conditions. The criteria listed under maintainability will also apply because of their relationship to reliability. Reliability is often expressed quantitatively as mean-time-between-failures (MTBF). The three questions raised by this term are:

1. What is a failure?
2. What causes a failure?
3. How is MTBF increased?

A failure may be defined as a deficiency which prevents a component or subassembly from performing its intended function under specified conditions.

The failure of a wheeled amphibian may be due to defective components, weak components and defective subassemblies or equipment, or environmental conditions such as rough seas, extremely steep slopes, etc., beyond the capabilities of the vehicle. No system can ever be assumed to be 100 percent reliable. The cost to design a wheeled amphibian which approached 100 percent reliability would be prohibitive.

The approach to improved reliability is:

a. Design for increased MTBF's

b. Subject the product under development or any developed component to prototype testing

c. Use component parts which have a record of reliability

To increase MTBF, there are two simultaneous approaches which are directly related to the design process. These are:

a. Design for ruggedness, i.e., use high safety factors and low working stresses, install oversized or overpowered drives and propulsion systems, etc.

b. Design with expendable parts which can be routinely replaced.

One of the tools available for achieving increased reliability is prototype testing which can provide knowledge of problem components before the vehicle is delivered to the operator. It will also provide the designer with an evaluation of his work and suggest areas for improvement. The ease of maintenance, or lack of it, will become obvious during tests. Testing can also provide information on which to base estimates of future MTBF's, especially if a detailed log of the vehicle's history is maintained. This log, if carried through the life of the vehicle, will be valuable in providing performance data to the designer.

Reliability components are selected from service experience. Performance data will be available for parts which have been used extensively, or tested by the Army or the manufacturer. Use of such components improves reliability because their known behavior can be designed into the system under consideration.

2-9.4 USE OF STANDARD MILITARY COMPONENTS

The major benefits to be gained from standardization are in the following areas—improved reliability, improved maintainability, reduced maintenance cost, reduced procurement cost, and reduced procurement time. However, standardization is not intended to hinder design improvement. Design improvements should take precedence over standardization whenever the improvement outweighs the advantages of standardization.

The contribution of standardization to improved reliability includes:

a. Use of standard items which are produced in large quantities insures a greater level of uniformity and refinement.

b. Use of standard items avoids long storage periods. Because of low demand, nonstandard items usually spend long periods in storage and are thus exposed to component deterioration.

c. Standardization promotes item familiarity, thereby minimizing the chances of misapplication, faulty handling, or faulty installation.

Standardization contributes to improved maintainability and reduced maintenance costs in the following ways:

a. Increased reliability reduces the frequency and extent of maintenance, thus reducing maintenance costs.

b. Maintenance downtime will be reduced since standard items will be more readily available and repairs will be more quickly effected.

c. Because of volume production, standard items will usually be less expensive than nonstandard items.

d. Costs associated with the logistic aspects of maintenance will be reduced since standardization allows greater interchangeability.

Standardization reduces the time and cost of system procurement because the development costs of nonstandard items are avoided. Even in cases where the nonstandard item is less expensive, this saving may be offset by the cost of introducing a new item into the supply system.

To summarize, the designer should attempt to use standard military components in his design for the following reasons:

a. Use of off-the-shelf items is cheaper than paying for the development of new items.

b. Components are readily available and have short delivery times.

c. Components have known properties and have more predictable reliability.

d. Use of known components will facilitate and speed maintenance because the components are familiar to personnel and the tools are available.

e. Use of known components will allow interchangeability between vehicles.

f. Components are part of an existing logistical support system.

Pertinent information concerning standard items which will help the designer are:

a. *Federal Cataloging Handbook*, H6-1.

b. *Numerical Index of Descriptive Patterns*, and other specifications and standards in accordance with MIL-STD-143.

c. *Army Repair Parts List*.

d. *Federal Supply Catalog* and AR 750-1.

2-10 HUMAN FACTORS

Human factors affect a wide variety of engineering requirements. Amphibians cannot operate without guidance and control by man. Hence, the designer must give due consideration to the physiological and psychological limitations of the human being. The operator and crew must be reasonably comfortable, have sufficient space in which to perform their functions, and have controls that are easily accessible and operable. More important, however, are the requirements for safe operation that include limits of human tolerance to vibration, noise, and shock.

Refs. 10, 11, and 12 discuss, in detail, the various aspects of human factor engineering which must be considered in developing satisfactory requirements.

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11. TM 21-62, *Manual of Standard Practice for Human Factors in Military Vehicles.*
12. HEL-STD-S-1-63, *Maximum Acceptable Noise Level for Army Materiel Command Equipment,* Human Engineering Laboratories, Aberdeen Proving Ground, Maryland, 1963.
13. AMCP 706-123, Engineering Design Handbook, *Hydraulic Fluids.*
14. AR 70-39, *Criteria for Air Transport and Airdrop of Materiel.*
15. MIL-STD-209, *Slinging Eyes and Attachments for Lifting and Tying Down Military Equipment.*
16. AMCP 706-115, Engineering Design Handbook, *Environmental Series, Part One, Basic Environmental Concepts*

CHAPTER 3 PAST AND PRESENTLY OPERATIONAL WHEELED AMPHIBIANS

SECTION I BRIEF HISTORY OF WHEELED AMPHIBIAN DEVELOPMENT

3-1 GENERAL REQUIREMENTS WHICH LED TO THE DEVELOPMENT OF RECENT AMPHIBIANS

Although the concept was not new, amphibians first came into prominence during WW II. In both the European and Pacific Theaters, the problem was the execution of assault and supply missions on unfriendly and (especially in the Pacific Theater) undeveloped shores. This led to the development of various amphibians. The need for logistic support under assault conditions led to the development of the type amphibian with which this handbook is concerned—the logistic support wheeled amphibian. The DUKW, the first operationally successful, mass-produced wheeled amphibian, grew out of the necessity to move unprecedented tonnages of supplies and equipment from ship to shore storage dumps over undeveloped beaches. The DUKW served a primary role in U.S. Army amphibious assault logistic operations throughout the war and for some time thereafter.

After WW II, developments in military equipment dictated that amphibians with a greater cargo delivery rate were required. This requirement stems from the great increase in fire power and mobility of the Armed Forces. Study of amphibian speeds and sizes required to provide the increased capability led to the realization that a "family" of amphibians was needed for flexibility in meeting various logistic problem demands.

A brief history of wheeled amphibian development follows, starting with the amphibious jeep and proceeding to the present.

3-2 THE AMPHIBIOUS JEEP

During the Autumn of 1941 the 1/4-ton, 4 X 4 general purpose truck was converted into an amphibian, the "amphibious jeep", for use in carrying personnel. The chassis, power plant, transmission, differential, and wheel components of the parent vehicle were used in a lightweight, welded hull. A propeller mounted in a tunnel, the necessary power take-off, a marine rudder, bilge pump, capstan, and other marine appurtenances were added to complete the conversion.

Over 12 thousand units were produced after limited field tests indicated the amphibious jeep had maximum speed of 65 mph on land and 6 mph in water. Insufficient testing of early production models, inadequate inspection and supervision of production, failure to provide for a continuing development program, failure to provide adequate training, and, particularly, failure to consult the using services before production began resulted in early rejection of the project as a technical and tactical failure. However, many lessons were learned from this unfortunate experience that were applied most profitably in the development of the DUKW.

3-3 THE DUKW

The DUKW was the first operationally effective wheeled logistic amphibian, although it was not used exclusively for supply functions. The basic development effort for this vehicle consisted of using the operationally proven GMC 6 X 6 truck and wrapping a watertight hull around it. This was an expedient approach, chosen because war time pressures did not permit extensive development and design time (spring of 1942).

The chassis of the GMC, CCKW 2½-ton, 6 X 6 truck was the basic structure and was enveloped in a welded steel barge type hull. The CCKW transmission, with five speeds forward and one reverse speed, was retained. The truck suspension system was retained, except that water seals were added. A marine power train—including screw propeller, shafting, power take-off, and a rudder—completed the major modifications. These changes gave the original 2½-ton truck the desired amphibious characteristics.

Following the initial introduction and production, certain design modifications were made to the DUKW to improve its waterborne capability. These included the following:

a. Originally, the cab was placed over the engine (forward) in the interest of operator visibility and maximum cargo space. Subsequently, the cab was moved just aft of the engine. This change increased the forward deck length, increasing operator protection and providing better engine accessibility. At the same time, the cargo compartment was widened in order to maintain available cargo space.

b. Water speed was increased from 5 mph to 6.5 mph. This was a result of modifications to the propeller, propeller location, engine carburetion, engine HP requirements, hull shape (adding 18 in. to the stern of the vehicle), and of providing covers for the front wheel housings.

The capability and adequacy of the DUKW, which was the result of a hasty but intensive design effort, is attested to by the fact that the DUKW did not become obsolete until nearly two decades after its introduction. The DUKW's were capable of transporting 5,000-lb payloads through plunging surf in excess of 10 ft and could attain land speeds up to 50 mph. The characteristics of the DUKW are given in Table 3-1 and Figure 3-1.

3-4 THE SUPERDUCK XM147

In the early 1950's, an improved version of the WW II DUKW was conceived, and was

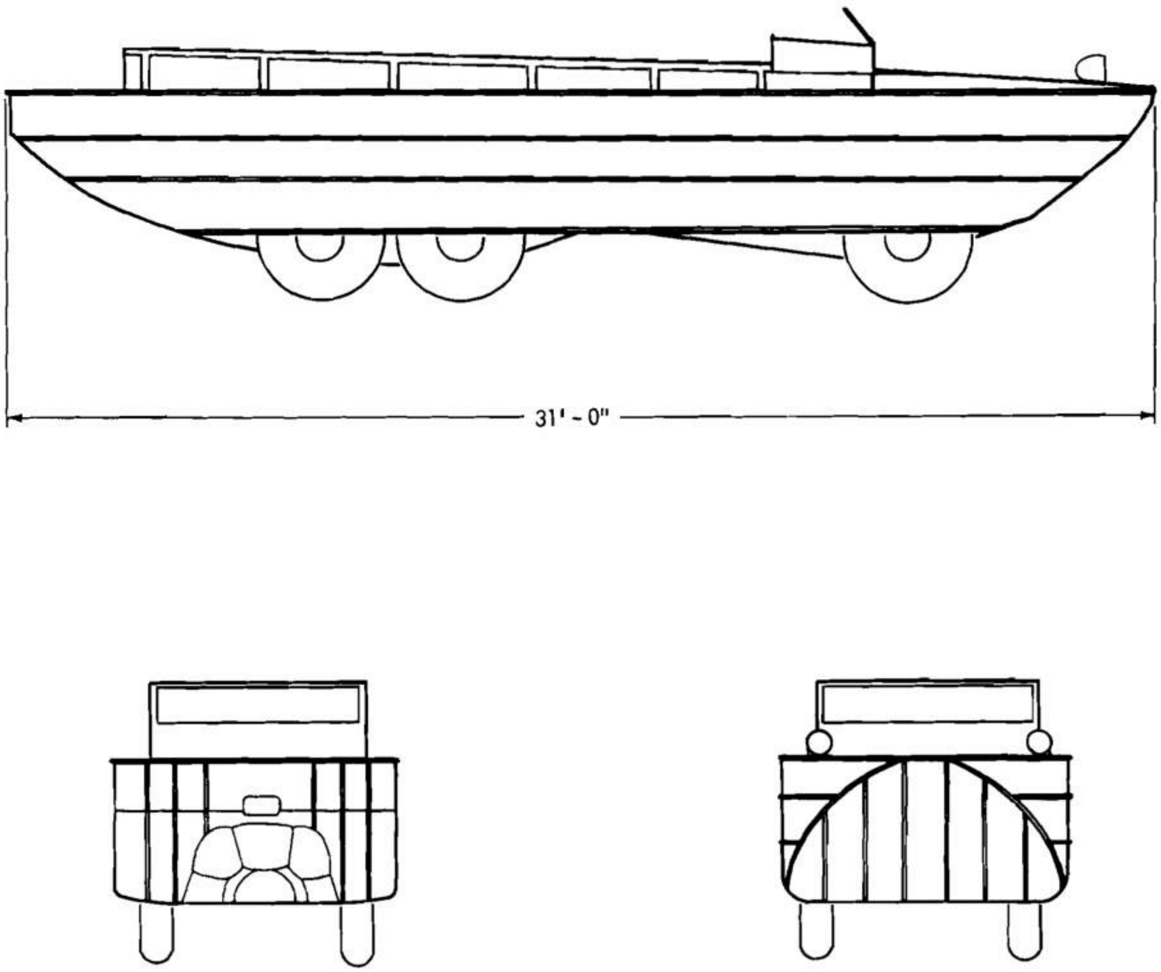


Figure 3-1. DUKW General Arrangement Sketch

named the SUPERDUCK. The first prototype produced was designated the XM147. It was tested in the surf off Monterey, California, in 1954. After brief testing, it was realized that certain major deficiencies existed which needed correction. This resulted in the development of a revised vehicle named the XM147E1 SUPERDUCK. This version included, as a major improvement over the DUKW and XM147, a prototype commercial torque converter and transmission. The XM147E1 employed a new power train, raised bow with surf plate, 10-ply desert-type nylon tires, and an attempt at improved marine steering. This vehicle was tested alongside the DUKW for comparison. The XM147E1 demonstrated superior land mobility, but was inferior to the DUKW in surf operations. Tests were prematurely concluded when the XM147E1 was swamped in heavy surf and severely damaged.

Another improved version of the SUPERDUCK, the XM147E2 was then developed. This amphibian employed an integral watertight hull and was provided in two modes, one with steerable propeller and one with rudders. During the tests, it was determined that this vehicle demonstrated inadequate engine cooling in the land mode of operation. It was also found that the transmission lacked adequate cooling and that the transfer cases were unsatisfactory for sustained marine services.

This led to the final revised version of the SUPERDUCK designated XM147E3, incorporating over 80 recommended changes based on test results of the XM147E2. This vehicle was tested in surf at Monterey, California, until it was swamped. A decision was then made to discontinue further consideration of the vehicle. The XM147E3 SUPERDUCK demonstrated only marginal improvement over the DUKW in most areas and needed major power train modifications in order to correct its deficiencies. The characteristics of the SUPERDUCK XM147E3 are given in Table 3-2 and Figure 3-2.

3-5 THE GULL XM148

During the period of SUPERDUCK evaluation, another experimental vehicle was introduced, designated the XM148 GULL. Its physical dimensions were approximately 15 percent greater than those of the DUKW. The GULL employed a molded bow, rather than the barge-type bows used on previous vehicles. Another innovation was the use of a glass-reinforced plastic hull. The GULL also had greatly increased power over all the previous wheeled amphibians, and featured unskirted wheel wells and an enclosed cab like the XM147. Vehicle characteristics are given in Table 3-3 and Fig. 3-3.

The vehicle prototype, as built, was considerably over the design weight. This overweight caused many components to fail which, otherwise, may have been satisfactorily designed. The GULL carried a very large winch with an inadequate foundation. Finally, the vehicle was plagued by failures in the glass fiber reinforced plastic hull. The state of the art in molding plastic hulls was in its infancy when the GULL was built. These problems, especially the weight problem, caused the project to end prematurely so that the GULL concept was never fully evaluated.

3-6 THE DRAKE XM157

The XM157 DRAKE was initially conceived as a 5-ton 6 X 6 vehicle similar to the XM147 SUPERDUCK. However, experience gained from the XM147 and XM148 prototypes led to changing the concept to an 8-ton, 8 X 8 amphibian having substantially increased marine capability.

The DRAKE used many components common to the SUPERDUCK and, in addition, had several unique features. It employed two independent power plant and drive train assemblies, each driving one forward and one rear axle. The power train consisted of a torque converter in series with six-speed gearing, a combination two-speed land-drive transfer case, and a two-speed marine propulsion power take-off. Dual trainable, retractable propellers driven through universal joints provided the marine propulsion and steering. Aluminum construction was used for both frame and hull. An air bellows suspension with independently sprung axle allowed partial retraction of the wheels and axles during waterborne operation.

Tests demonstrated that the DRAKE was superior in land mobility to the XM147 and DUKW. Its air suspension system provided good

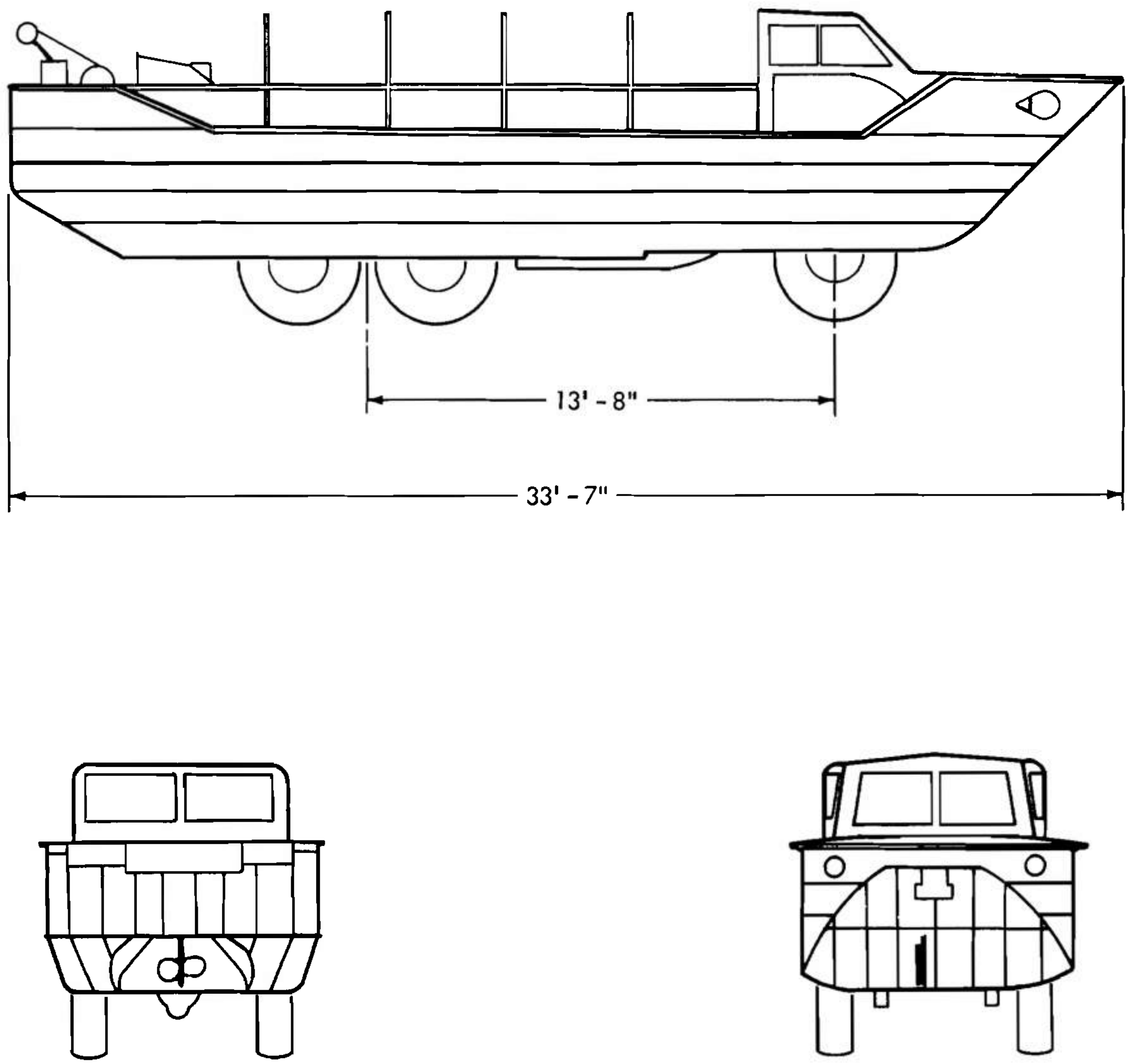


Figure 3-2. SUPERDUCK General Arrangement Sketch

lateral stability, and control was facilitated by power steering. The central tire inflation system also contributed to this mobility. In the waterborne mode, it demonstrated maneuverability superior to that of previous amphibians. However, there was a marked directional instability in the waterborne mode. It also proved capable of higher waterborne speeds than the previous amphibians. Vehicle characteristics are given in Table 3-4 and Fig. 3-4.

The greatest deficiency in the DRAKE design was that some bearings were underdesigned resulting in numerous breakdowns in the differential. The vehicle was also poorly designed from a maintenance standpoint. To service the engine, it was necessary to remove the cab. Since there was no major improvement in performance over the SUPERDUCK, the concept was abandoned.

3-7 THE PRESENT FAMILY OF OPERATIONAL LOGISTIC WHEELED AMPHIBIANS

The present family of amphibians consists of 3 vehicles—the LARC LX, the LARC V, and the LARC XV, which are capable of handling 60

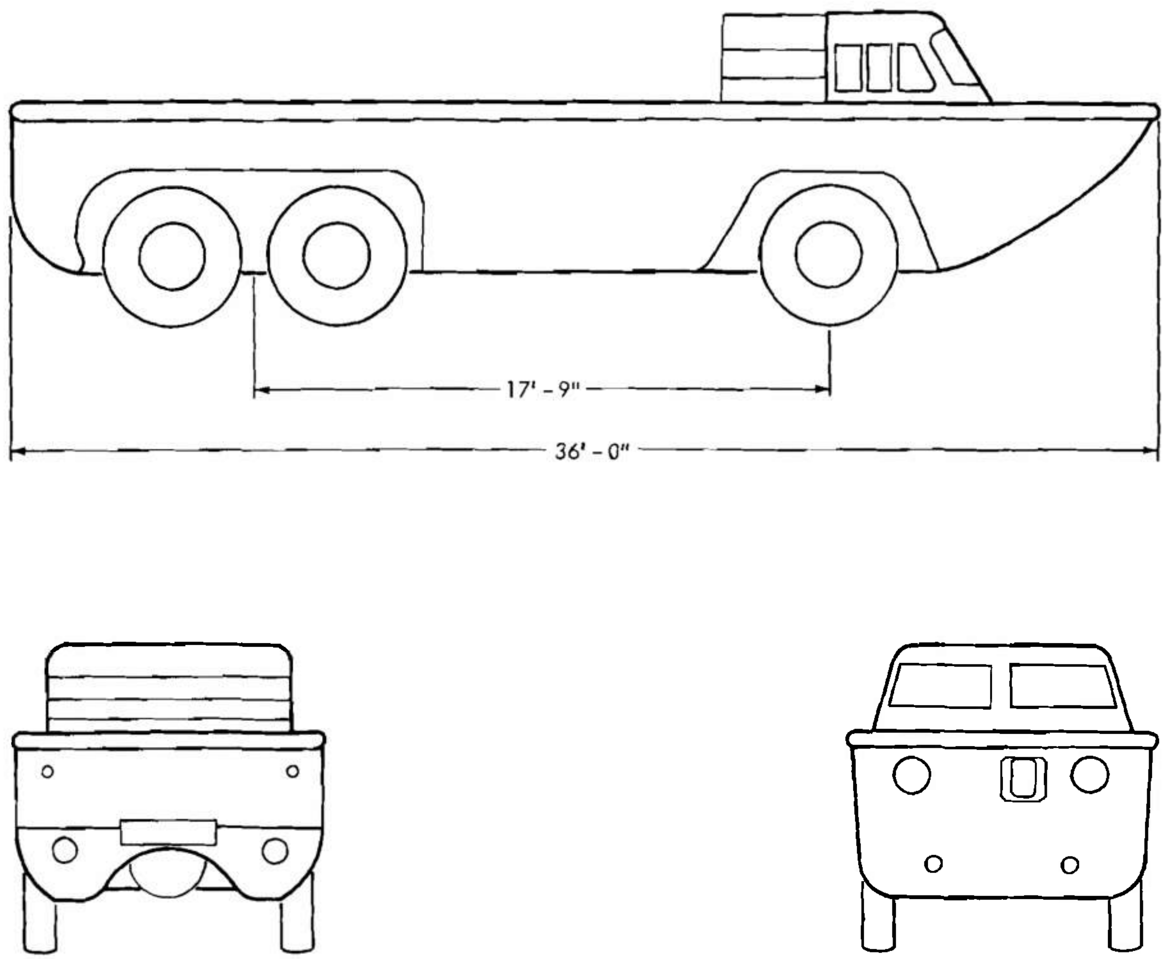


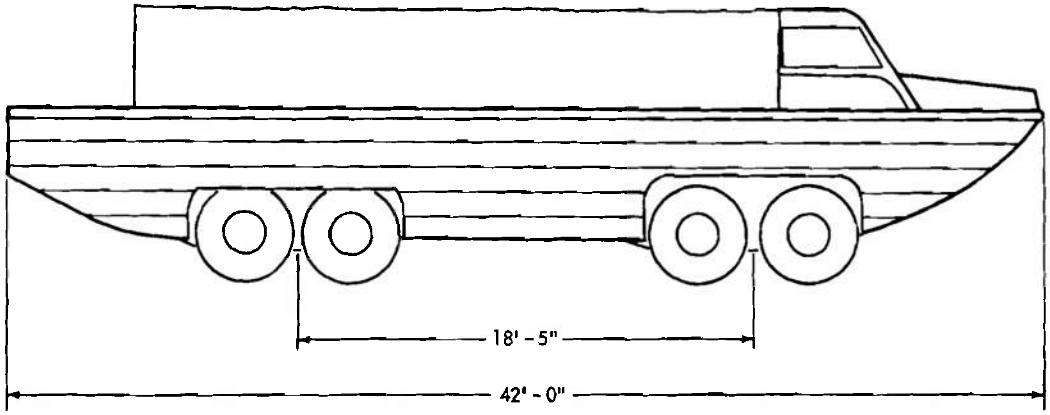
Figure 3-3. GULL XM148 General Arrangement Sketch

tons, 5 tons, and 15 tons of payload, respectively. It is fitting to refer to these three amphibians as a family since they complement one another in fulfilling the overall logistic requirements for amphibious operations. There exists a concept of usage for these amphibians and each one fulfills a part. The LARC V will transport nonmobile general cargo and has maximum roadability. The LARC XV will transport nonmobile general cargo (it can carry three loaded CONEX containers), lighter types of mobile equipment, and has good roadability. The LARC LX will carry all the heavy mobile and nonmobile equipment and has limited roadability. The LARC V and LARC XV will transport 75 percent of all the dry cargo, general cargo, special cargo, ammunition, and other Class IV wheeled and tracked vehicles, which are used by the military services. The LARC LX will transport the remaining 25 percent. A description of each vehicle follows.

3-7.1 THE LARC LX

The first of the family to be developed was the LARC LX. In fact, the LARC LX concept was initiated quite some time before the other LARC's. Development of this amphibian was initiated in 1951 and was carried forward during the same period of time as the SUPERDUCK development.

The LARC LX was developed to meet requirements for an amphibious vehicle of large size to transport heavy and bulky equipment from ship to shore and across unimproved beaches. The LARC LX is a Diesel-powered,



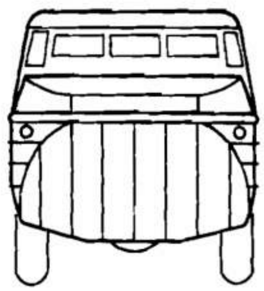


Figure 3-4. DRAKE XM157 General Arrangement Sketch

rubber-tired, amphibious lighter, with a bow loading ramp, center cargo compartment, closed stern, and wing machinery spaces. The basic structure is of welded steel. Four 6-cylinder GM Detroit Diesel engines are used for the propulsion power plant.

For marine propulsion, the two engines on the port side are connected end to end by a fluid coupling and shaft joined to a gathering box. The box is connected via a right angle drive to the reverse and reduction marine gear. Power is then transmitted from the marine gear, through shafting, to a propeller. The starboard installation is identical.

The land drive is so arranged that each of the four wheels is independently driven from one of the Diesel engines. A torque converter and transmission are mounted at one end of each engine. Each transmission is coupled to a miter box by a spheriflex coupling. The miter box effects the necessary right angle drive to the wheel column. The wheel column in turn transmits the power through a coupling and a final drive unit to a set of planetary gears in the wheel hub. Braking is accomplished by restraining the drive shafts, using air brakes attached at the miter box.

Wheel steering is accomplished by the use of tiller arms splined to the top of each wheel column, and attached to two hydraulic rams which provide the steering power. Marine steering is accomplished by the use of a wire rope loop which is attached to the starboard front wheel steering arm and to the quadrant which controls the two welded steel rudders. The characteristics of the LARC LX are given in Table 3-5 and Fig. 3-5.

3-7.2 THE LARC V

The LARC V was the first true production replacement for the WW II DUKW. Unlike the DUKW which, for the sake of expediency, was designed more or less as a floatable truck, the

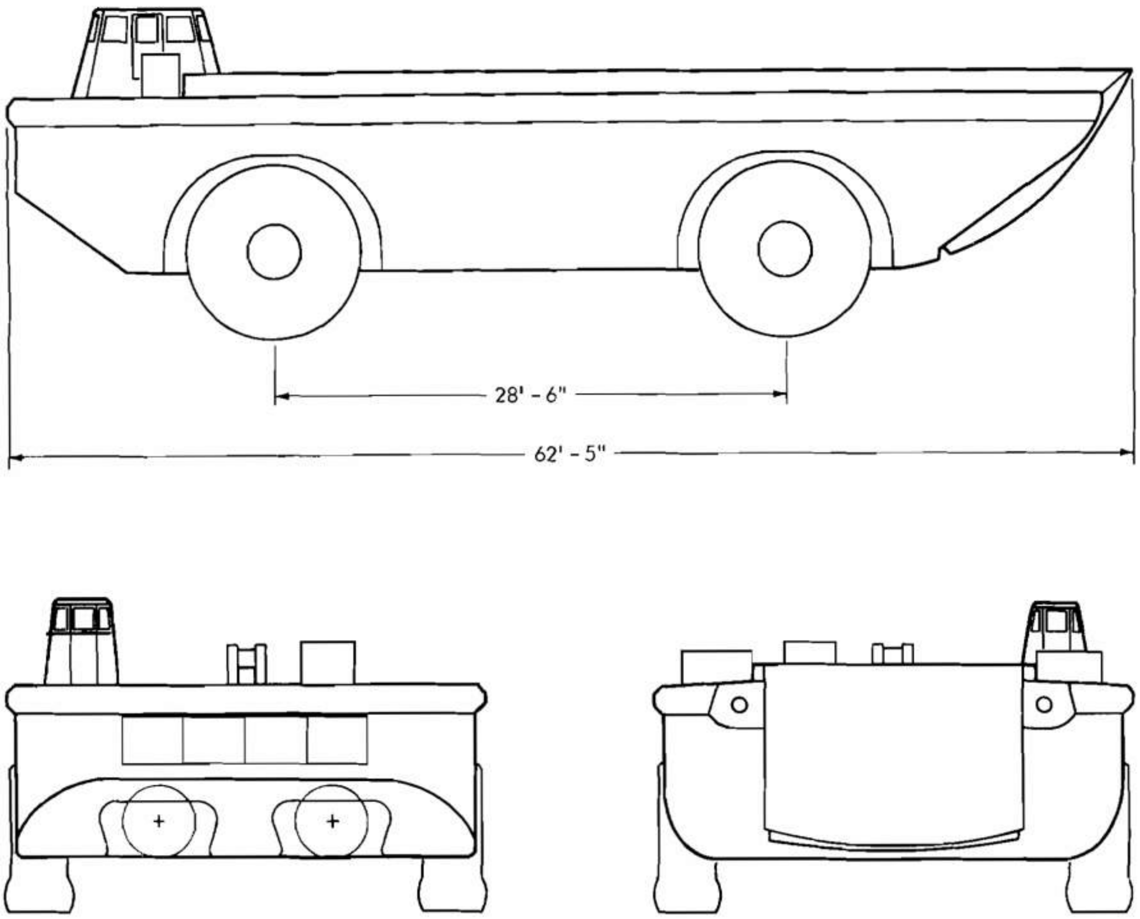


Figure 3-5. LARC LX General Arrangement Sketch

LARC V was designed from the ground up as a true amphibian. Its mission is to transport nonmobile general cargo from ship to shore, over undeveloped beaches and roads. The LARC V has been in service and classified Standard A since 1960 and has been in troop unit service since 1963.

The production model of the LARC V is a wheeled amphibian powered by a 300-horsepower, four-cycle, 8-cylinder, Cummins Diesel engine providing land propulsion through 2 or 4 driven, rubber-tired wheels, and waterborne propulsion through a single propeller. The width was increased by 1.0 ft during development to satisfy stability requirements. The engine, which is located in the stern of the amphibian (over the propeller tunnel), drives forward to a centrally located transfer differential transmission, where power is divided for driving the wheels and/or the propeller. The amphibian is equipped with 18 X 25 in., 12-ply, off-road type, low-pressure, tubeless tires, which also serve as the suspension system. Land-mode steering is provided using the bow wheels. Marine steering is provided through cables attached to the land steering linkage. The rudder and the front wheels operate together. The single propeller operates in a tunnel and is fitted with a Kort nozzle. The designed speed is 10 mph in water and 30 mph over land under full load conditions. The hull is constructed of aluminum.

The effectiveness of the LARC V, due to its increase in both cargo capacity and water speed, can best be expressed by comparison with other amphibians. Based on evaluation tests, the LARC V is the equivalent of 2.4 DUKW's or 1.6 SUPERDUCK's. The LARC V has demonstrated

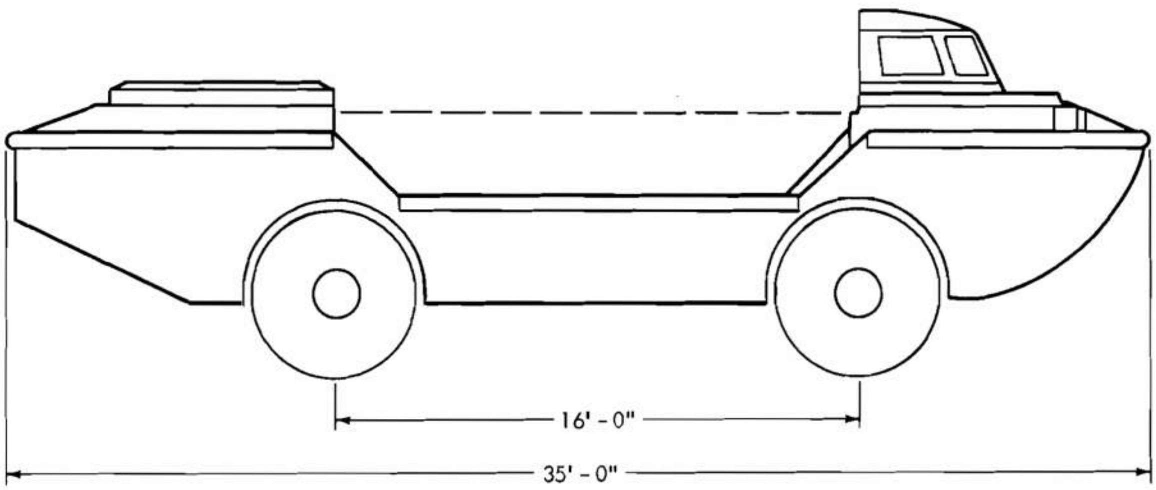
superiority to the SUPERDUCK in land mobility, maintainability, and design simplification; it is also a lighter vehicle. The vehicle characteristics are given in Table 3-6 and Fig. 3-6.

3-7.3 THE LARC XV

Since the LARC LX was developed to fulfill requirements for transport of bulky, heavy cargo, and the LARC V for transport of small general cargo; the LARC XV was developed to meet requirements for a medium self-propelled lighter. The mission of the LARC XV is to transport nonmobile general cargo and lighter types of mobile equipment from ship to shore, over the beach, and over the roads. The LARC XV is designed for expedient unloading using a fork lift truck for unloading nonmobile cargo, and using a ramp for mobile cargo.

The LARC XV is powered by two 4-cycle, 8-cylinder, V-type Diesel engines, the same as are used in the LARC V. The vehicle is driven on land through four large, low-pressure tires which also serve as the suspension system. As with the LARC V, the beam was increased, during development, by 2.0 ft for stability. In water, it is propelled by a single 4-bladed propeller. Steering is hydraulic; 2-wheel, 4-wheel, and oblique steering are possible. Marine steering is accomplished by a 3-bladed box rudder which is actuated by the same hydraulic system which provides for land-mode steering. Wheels and rudder turn together. The LARC XV hull is constructed of aluminum.

A unique feature of the LARC XV is that it is



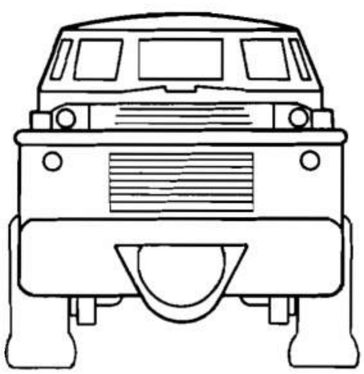


Figure 3-6. LARC V General Arrangement Sketch

a two-way vehicle. In the waterborne mode, it is operated in the direction which places the boat-shaped bow forward and the cab aft, thus providing good operator protection. In the land mode, it is operated in the opposite direction, with the cab forward and the bow to the rear, thus providing good visibility for the operator. LARC XV characteristics are given in Table 3-7 and Fig. 3-7.

Fig. 3-8 shows the LARC family of amphibians and Fig. 3-9 shows the DUKW, SUPERDUCK, and prototype LARC V.

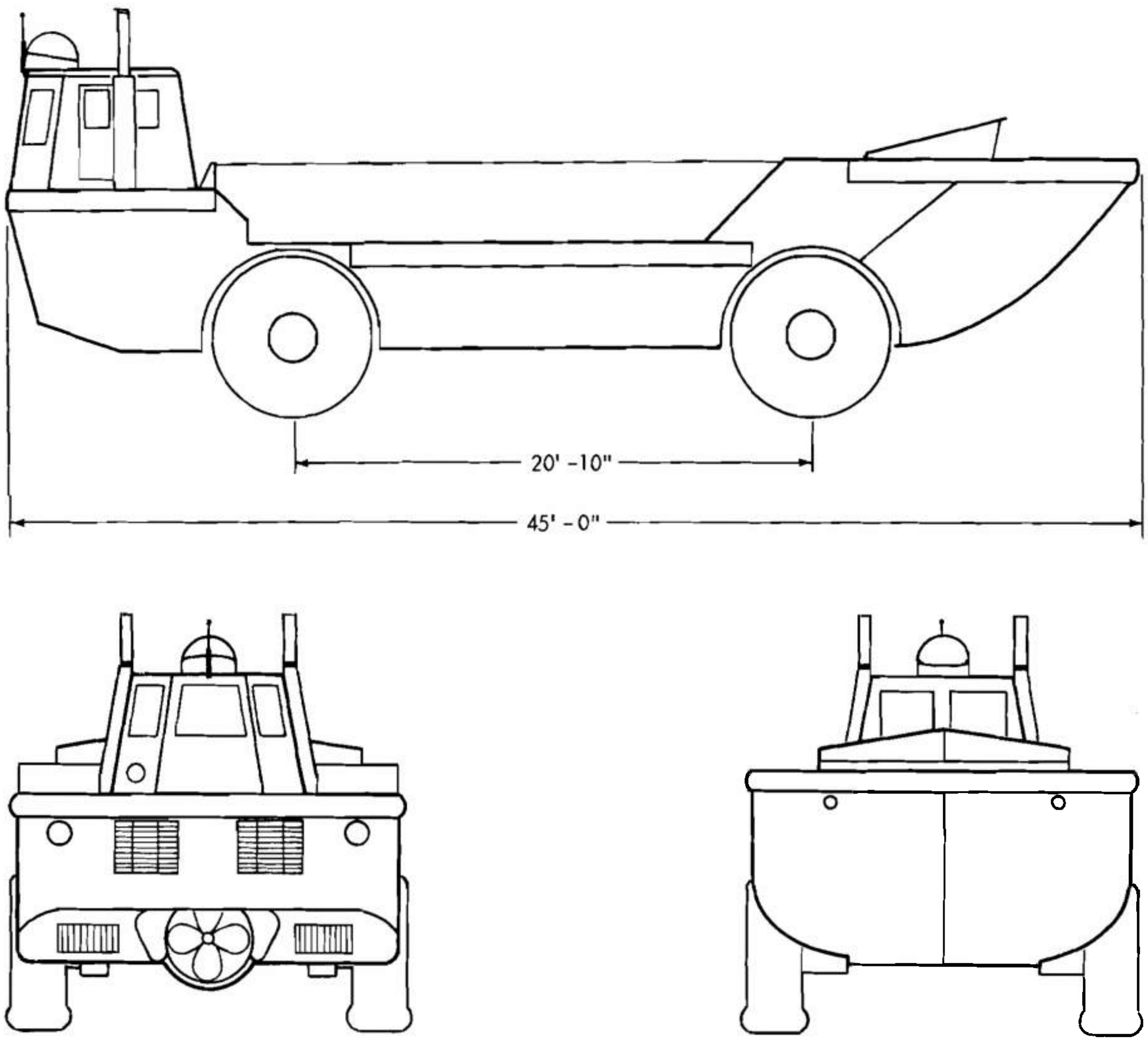


Figure 3-7. LARC XV General Arrangement Sketch

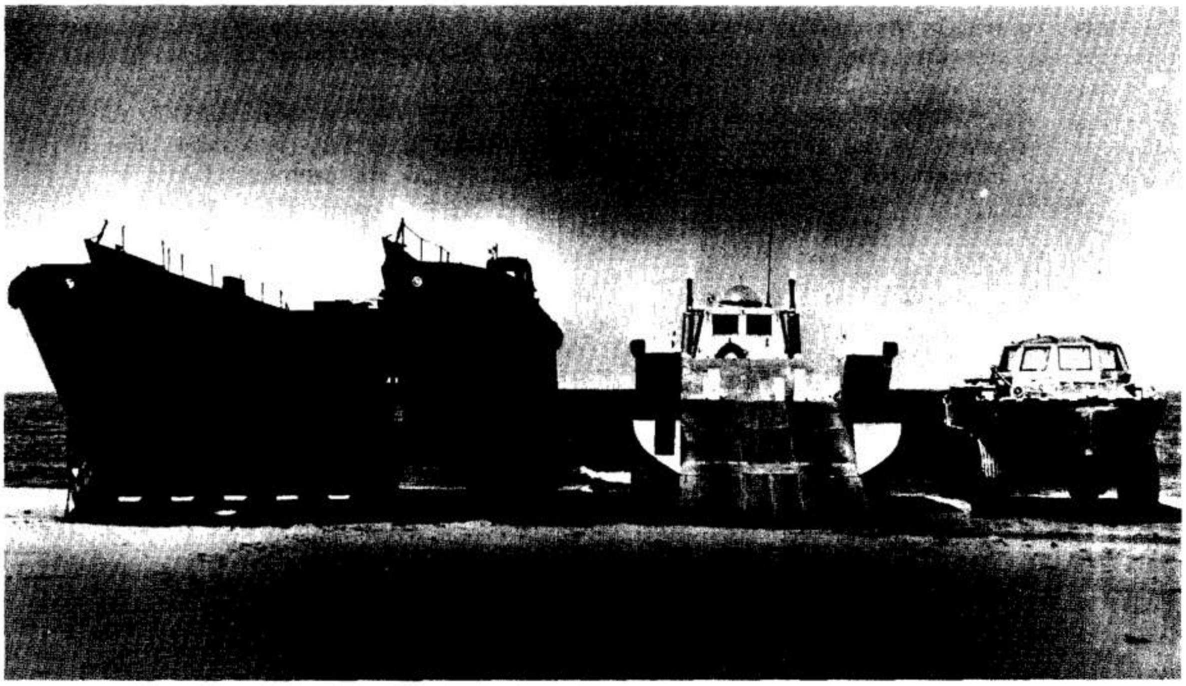


Figure 3-8. LARC Amphibians: LARC LX, LARC XV, and LARC V

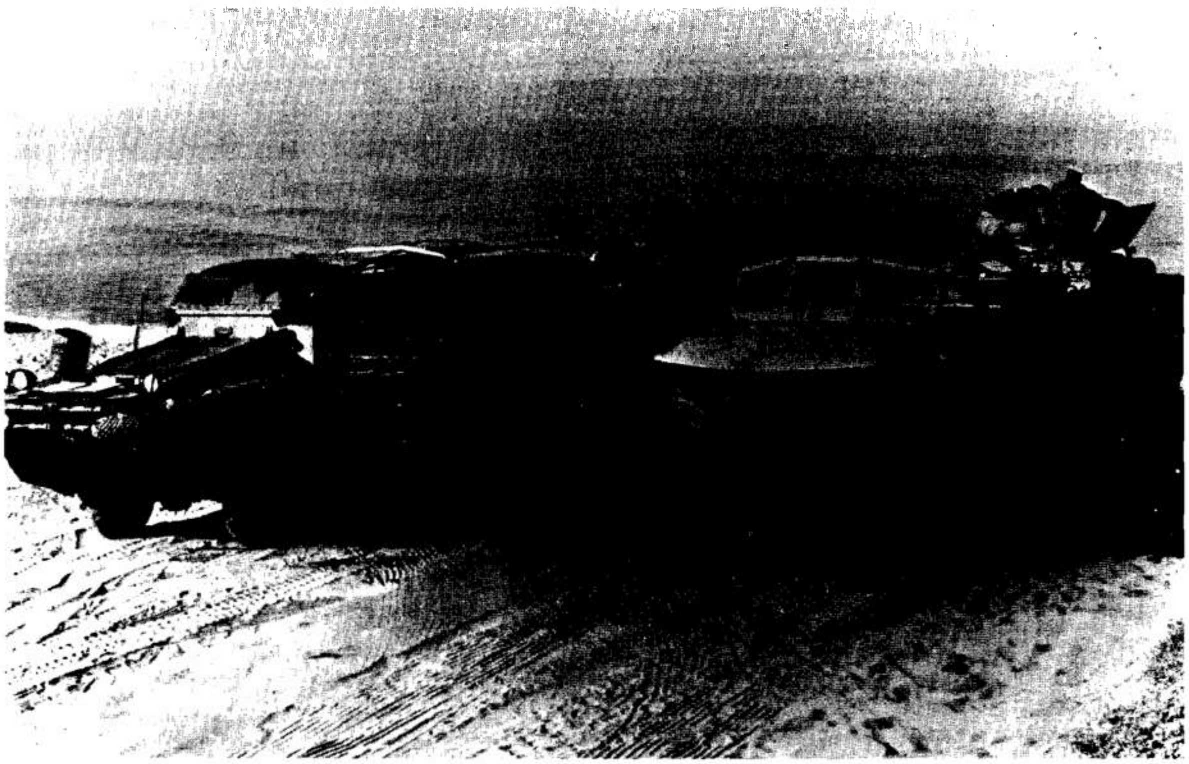


Figure 3-9. WW II DUKW, SUPERDUCK, and Prototype of the LARC V

SECTION II TABULATION OF VEHICLE CHARACTERISTICS

3-8 VEHICLE CHARACTERISTICS OF PAST AND PRESENTLY OPERATIONAL WHEELED AMPHIBIANS

TABLE 3-1 DUKW–VEHICLE CHARACTERISTICS

Manufacturer	General Motors Corp.
Width, ft-in.	8-2-7/8
Length, ft-in.	31-0
Height, ft-in.	9-2½
Min Ground Clearance, in.	11¼
Freeboard, in.	24 fwd; 16 aft
Draft, ft-in.	3-6 fwd; 4-3 aft
Wheelbase, ft-in.	13-8
Track Width, ft-in.	5-3-5/8
Angle of Approach, deg	38
Angle of Departure, deg	25
Break Angle, deg	15
Land Speed, mph	50
Water Speed, mph	6.2
Land Range, mi	240
Water Range, mi	35 at 6 mph
Turn Radius (Land), ft	35
Turn Radius (Water), ft	20
Cargo Capacity	5,175 lb or 25 troops
Cargo Space (L×W×H), ft-in.	12-4 × 6-10 × 2-4
Total Weight, lb	20,055
Grade Ability, %	60
Surf Ability, ft	10
Engine Data	GMC Model 270 6-cyl gasoline engine 93 BHP at 2750 rpm
Fuel Type	Gasoline 72-80 octane
Marine Propulsor	3-bladed propeller; D = 25 in.; P = 14 in.
Suspension System	Truck type; leaf springs; 6 × 6
Brake System	Hydraulic
Tire Size	11 × 18 in. 10-ply
Land Steering	2-wheel, mechanical
Marine Steering	Rudder
Crew Size	2
Hull Material	Steel
Central Tire Inflation System	

TABLE 3-2 SUPERDUCK (XM147E3)–VEHICLE CHARACTERISTICS

Manufacturer	General Motors Corp.
Width, ft-in.	9-3/8
Length, ft-in.	33-7
Height, ft-in.	9-4
Min Ground Clearance, in.	13
Freeboard, in.	35 fwd; 28 aft
Wheelbase, ft-in.	13-8
Track Width, ft-in.	5-11¾
Angle of Approach, deg	40
Angle of Departure, deg	22
Land Speed, mph	48
Water Speed, mph	7
Turn Radius (Land), ft	35
Turn Radius (Water), ft	25
Cargo Capacity, lb	8,000
Cargo Space (LXWXH), ft-in.	15-2 X 7-4 X 2-10
Total Weight, lb	27,780
Grade Ability, %	60 at 1.64 mph
Surf Ability, ft	10
Engine Data	GMC 302-55; 6-cyl gasoline engine, 155 hp
Fuel Type	Gasoline
Fuel Capacity, gal	108
Marine Propulsor	3-bladed propeller; D = 31 in., P = 25 in.
Suspension System	Truck Type, Leaf Spring; 6 X 6
Brake System	Air over hydraulic, disc type
Tire Size	12.5 X 20 in.; 12-ply
Land Steering	Recirculating ball type, 2-wheel, power-assisted
Marine Steering	Single Rudder
Electrical System	24-V DC
Crew Size	2
Hull Material	Welded Steel

TABLE 3-3 GULL (XM148)–VEHICLE CHARACTERISTICS

Manufacturer	ACF Brill
Width, ft-in.	5-9
Length, ft-in.	36-0
Height, ft-in.	9-10½
Min Ground Clearance, in.	20-5/8
Wheelbase, ft-in.	17-9
Angle of Approach, deg	30
Angle of Departure, deg	30
Land Speed, mph	63
Water Speed, mph	8
Land Range, mi	360
Water Range, mi	66
Cargo Capacity, lb	10,000
Cargo Space (LXW), ft-in.	16-5 X 8-4
Engine Data	Hall-Scott Model 485; 6-cyl; 300 BHP
Fuel Type	Gasoline
Fuel Capacity, gal	140
Suspension System	6 X 6
Land Steering	2-wheel
Hull Material	Glass-reinforced plastic

TABLE 3-4 DRAKE (XM157)–VEHICLE CHARACTERISTICS

Manufacturer	General Motors Corp.
Width, ft-in.	42-0
Length, ft-in.	10-0
Height, ft-in.	10-10
Min Ground Clearance, in.	18
Freeboard, in.	28
Wheelbase, ft-in.	18-5
Angle of Approach, deg	34
Angle of Departure, deg	23
Land Speed, mph	44
Water Speed, mph	9
Land Range, mi	396
Turn Radius (Land), ft	49
Turn Radius (Water), ft	22
Cargo Capacity, lb	16,000
Cargo Space (L×W×H), ft-in.	23-0 × 8-11 × 3-3
Total weight, lb	49,200
Grade Ability, %	60
Engine Data	(2) GMC, Model 302, 155 hp
Fuel Type	Gasoline
Fuel Capacity, gal	240
Marine Propulsor	(2) Steerable propellers; D = 31 in.; P = 24 in.
Suspension System	Air bellows type; 8 × 8; partial retraction
Brake System	1 (i.e., no redundancy)
Tire Size	14.75 × 20 in.; 10-ply
Land Steering	4-wheel (front)
Marine Steering	Directed thrust (trainable propeller)
Hull Material	Welded aluminum
Central Tire Inflation System	

TABLE 3-5 LARC LX—VEHICLE CHARACTERISTICS

Manufacturer/Designer	Treadway/Pacific Car and Foundry
Width, ft-in.	26-11
Length, ft-in.	62-5
Height, ft-in.	19-7 reducible to 14-7 for shipping
Min Ground Clearance, in.	30
Freeboard, ft-in.	5-0 fwd; 4-6 aft
Draft, ft-in.	8-2 fwd; 8-8 aft
Wheelbase, ft-in.	28-6
Track Width, ft-in.	23-2
Angle of Approach, deg	28
Angle of Departure, deg	28
Break Angle, deg	13½
Land Speed, mph	17 (loaded); 20.5 (empty)
Water Speed, mph	7 (Loaded); 7.5 (empty)
Land Range, mi	300 at 10 mph
Water Range, mi	150
Turn Radius (Land), ft	75
Cargo Capacity, lb	120,000
Cargo Space (L×W×H), ft-in.	41-11 × 14 × (6-2 fwd; 4-6 aft)
Total Weight, lb	319,000
Grade Ability, %	60
Side Slope, %	25
Surf Ability, ft	14
Engine Data	(4) GM Series 6-71 Diesel; rated at 165 BHP at 2100 rpm; separate drive for each wheel
Fuel Type	Marine Diesel
Fuel Capacity, gal	600
Marine Propulsor	(2) 3-bladed propellers; D = 48 in., P = 29 to 32 in.
Suspension System	Rigid, 4 × 4
Brake System	Air type, B. F. Goodrich Model H-500-1
Tire Size	36 × 41 in., 48-ply rating
Land Steering	Hydraulic selective 2-wheel, 4-wheel, or oblique
Marine Steering	Twin rudders
Electrical System	24-V DC single-wire
Hydraulic System, gal	300
Crew Size	4-5
Hull Material	Welded steel
Unit Cost	$390,000 (6/68)

TABLE 3-6 LARC V—VEHICLE CHARACTERISTICS

Manufacturer/Designer	Consolidated Diesel Electric Corporation/Ingersoll-Kalamazoo Div. of Borg-Warner
Width, ft-in.	10-0
Length, ft-in.	35-0
Height, ft-in.	10-2
Min Ground Clearance, in.	16
Freeboard, in.	10
Draft, ft-in.	4-3
Wheelbase, ft-in.	16-0
Track Width, ft-in.	8-5
Angle of Approach, deg	20.7
Angle of Departure, deg	27
Break Angle, deg	15
Land Speed, mph	28
Water Speed, mph	8.6
Land Range, mi	200
Water Range, mi	25
Turn Radius (Land), ft	36.5
Turn Radius (Water), ft	77 clockwise, 59 counter-clockwise
Cargo Capacity	10,000 lb or 20 troops
Cargo Space (LXWXH), ft-in.	16-0 X 8-8 X 2-6
Total Weight, lb	30, 950
Grade Ability, %	60 at 2.41 mph
Side Slope, %	20
Surf Ability, ft	10
Engine Data	(1) Cummins 8-cyl, 4-cycle Diesel engine; 300 BHP at 3000 rpm
Fuel Type	Marine Diesel
Fuel Capacity, gal	144
Marine Propulsor	(1) 3-bladed shrouded propeller; D = 30 in., P = 30 in.
Suspension System	Rigid, 4 X 4
Brake System	Hydroretarder and hydraulic disc on transmission shaft
Tire Size	18 X 25 in; 12-ply
Land Steering	Power assist—2-wheel
Marine Steering	Single rudder and front wheels
Electrical System	24-V DC single-wire
Hydraulic System, gal	19
Crew Size	2
Hull Material	Welded aluminum
Bollard Pull, lb	3,260
Unit Cost	$44,200 (6/68)

TABLE 3-7 LARC XV—VEHICLE CHARACTERISTICS

Manufacturer/Designer	Freuhauf, Inc./Ingersoll-Kalamazoo Div. of Borg-Warner
Width, ft-in.	14-6
Length, ft-in.	45-0
Height, ft-in.	13-8
Min Ground Clearance, in.	16½ (to propeller shroud)
Freeboard, in.	15
Draft, ft-in.	5-1
Wheelbase, ft-in.	20-10½
Track Width, ft-in.	12-6
Angle of Approach, deg	22½
Angle of Departure, deg	30½
Break Angle, deg	13½
Land Speed, mph	29.9
Water Speed, mph	9.5
Land Range, mi	200
Water Range, mi	120
Turn Radius (Land), ft	55
Turn Radius (Water), ft	77
Cargo Capacity	30,000 lb or 53 troops
Cargo Space (L×W×H), ft-in.	24-0 × 13-6 × 3-3
Total Weight, lb	75,200
Grade Ability, %	40 at 1 mph
Side Slope, %	25
Surf Ability, ft	12
Engine Data	(2) Cummins V-8 Diesel engine; 4-cycle; 300 BHP at 3,000 rpm
Fuel Type	Marine Diesel
Fuel Capacity, gal	360
Marine Propulsor	(1) 4-bladed propeller with modified Kort nozzle; P = 36 in., D = 34 in.
Brake System	(4) Hydraulic disc
Tire Size	24 × 29 in.; 16-ply rating
Land Steering	Selective 2-wheel, 4-wheel, or oblique; hydraulic power
Marine Steering	3-blade box rudder and wheels
Electrical System	24-V DC single-wire
Hydraulic System, gal	25; 1,900 psi closed center
Crew Size	2
Hull Material	Welded aluminum
Bollard Pull, lb	7,700
Unit Cost	$165,000 (6/68)

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PART TWO DESIGN OF WHEELED AMPHIBIANS

CHAPTER 4 INTRODUCTION TO THE DESIGN PROCESS

4-1 PHASES OF THE DESIGN PROCESS

The design of a wheeled amphibian may be considered to take place in four phases. In the course of these phases the design progresses from general considerations, such as size and power, to specific considerations including the design, selection and arrangement of components and, finally, testing and evaluation. The four phases of the design process are:

1. Conceptual Design. During this phase the basic characteristics of the candidate amphibian are developed. This design work is done in support of system studies and used to develop realistic and specific requirements.
2. Preliminary Design. During this phase the characteristics developed in the conceptual design are checked and modified until the requirements are satisfied. This is done by designing, selecting, and arranging the major components and systems of the amphibian. This includes the necessary major hardware trade-offs to produce a cost-effective vehicle.
3. Detail or Final Design. During this phase the preliminary design is developed into a set of final plans from which the amphibian can be built. This involves the design, selection, and arrangement of all components and systems in the amphibian.
4. Testing and Evaluation. During this phase prototype vehicles are subjected to the tests necessary to show that the design satisfies the requirements. Testing and evaluation provide feedback on the validity of the design procedures used, and define the defects and problems that must be overcome for the amphibian to meet the requirements. Later in the life cycle, testing is carried out as part of product improvement programs.

These four phases of the design process are known by a variety of names. In general there is no sharp break between the different design phases; rather, they tend to overlap. This is especially true for the conceptual and preliminary design phases.

It is important to realize that, in general, the successive design phases of a wheeled amphibian will not be carried out by the same group. In the past the conceptual and preliminary designs have been conducted by designers working directly for the U. S. Army. The final design and some testing have been carried out by the contractor who builds the prototype vehicle. The remainder of the testing is conducted by the U. S. Army Test and Evaluation Command (USATECOM). This division of the design process introduces problems in control and information flow in the design. These areas are covered in pars. 4-2 and 4-3. However, design could also be obtained under the "Total Package Procurement" concept. In addition, design and other phases in a weapon system's life cycle could be a function of a Department of Defense or Department of Army Project Manager, or an AMC Product Manager. Project/Product Management of an equipment at any period during its life cycle vests one individual with ultimate responsibility for all aspects of the item he manages. The Project/Product Manager cuts across all functional lines engineering, logistics, programming and procurement—thereby greatly improving the vertical and horizontal coordination between these activities. The concept of project/product management is presented in Refs. 1 through 4.

4-2 CONTROL OF THE DESIGN PROCESS

In order to produce a successful wheeled amphibian in a reasonable time, the design process must be under the direction of a central authority. The person or persons who act as this authority must have the technical ability and authority to direct and control the design development. They must also be responsible for the design satisfying the requirements. Experience in the design of many different items has shown that when the responsibility for the satisfactory performance of the end product is clearly defined, the chances of success early in

the project are greatly increased. This problem and, in particular, the need to establish responsibility through quantitative testable end requirements are covered in detail in Ref. 5.

As indicated in par. 4-1, the entire design process will probably not be carried out by one group. As a result, those with the ultimate authority and responsibility must plan for the necessary supervision, schedules, information, and coordination. In particular, time can be saved if there is the proper coordination among the final design, prototype construction, and testing. This includes the formulation of test teams, approval of regulatory bodies, the development of engineering test programs, and the correction of problems uncovered in tests.

4-3 INFORMATION FLOW IN THE DESIGN PROCESS

The design of a wheeled amphibian is, by its nature, an iterative procedure. Assumptions are made, then checked by calculations and layouts, and then modified as necessary. These iterations are repeated until a satisfactory design has been developed. The actual iteration cycles used are presented in Chapters 5 and 6.

Because of its iterative nature, the success of the design process depends on the timely flow of information. The more the design work is speeded up by concurrent work on different parts of the design, the more critical the problem becomes. The information flow in a design is both vertical from phase to phase and horizontal within a given phase.

If different design phases are carried out by different groups of designers, vertical information flow becomes important. In addition to the usual information transferred between phases—such as requirements, plans, and calculations—other information is useful. This includes the assumptions made, the important technical decisions, the reasons for these decisions, and, if appropriate, a brief description of the design philosophy.

Horizontal information flow becomes more important as the number of designers working on a design increases and is thus most important during final or detail design. This information is used to coordinate concurrent work on different parts of the design. The need for coordination among designers working such components as the hull, running gear, and drive train is obvious.

4-4 ORGANIZATION OF DESIGN CHAPTERS

The next four chapters of this handbook—organized by phases of the design process—present wheeled amphibian design procedures, techniques, and data. Chapter 5 covers conceptual design; Chapter 6 covers preliminary design; Chapter 7, detail design; and Chapter 8, testing and evaluation. Each of these chapters presents the procedures, techniques, and data appropriate to the particular design phase. There is some repetition of topics between chapters since the same topics must be considered in more than one phase of the design.

In effect, Chapters 5 through 8 form an interrelated group and should be considered as such. In order to understand the proper use of some of the general design procedures and data presented in Chapters 5 and 6, it will be necessary to review the basic concepts presented in Chapter 7. This is especially the case with respect to waterborne resistance, propulsion, and stability.

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CHAPTER 5 PARAMETRIC AND CONCEPTUAL DESIGN STUDIES

5-1 OBJECTIVE

The basic objective of conceptual design studies of wheeled amphibians is to define basic technical and performance characteristics for use in evaluation studies and for input to following design phases. Parametric design studies involve the development of a number of conceptual designs in which the inputs are varied systematically to determine design interrelationships, trade-offs, and optimum characteristics. These studies must be carried out rapidly and in sufficient detail to identify the gross characteristics of the craft, and to establish feasibility and technical problems areas.

5-1.1 BASIC PURPOSE

A parametric design study or a conceptual design may be carried out for a number of reasons, including the following:

a. Trade-off and Optimization Studies. In the early stages of the design process it is necessary to choose an optimum combination of characteristics from the many possible alternatives. Included in these alternatives are variables such as dimensions; hull material; and selection of engine, propulsor, and drive train. These selections are made from trade-off studies which involve the development of conceptual designs based on the possible alternatives. The choice is then made among the possibilities on the basis of a selection criterion which is often in the form of a cost-effectiveness ratio. Ref. 1 presents a detailed discussion of the use of conceptual designs in this type of optimization study. The information and data developed in such studies form the inputs to the preliminary design phase.

b. Requirement Development. The military or performance characteristics for a wheeled amphibian will be given in a requirement or specification such as a QMR (Qualitative Materiel Requirement) or SDR (Small Development Requirement). The initial form of this type of requirement comprises the input to the start of the conceptual design phase. In their final form these requirements must be free of impossible conflicts and lead to a practical vehicle if a great deal of time and effort is not to be wasted. The recommended approach to achieve this goal is the development of conceptual designs which include all of the requirements. This procedure will confirm that it is feasible to meet the requirements and facilitate a realistic estimate of the scope of effort required.

c. Background or Input Data. System analysis studies on military systems may be conducted by or for elements of the United States Government that make use of wheeled amphibians. The amphibian designer will be asked to provide technical data on the selection of wheeled amphibians for use in these systems. In many cases these systems will be intended for a future date and it will be necessary to develop information on vehicles that can be designed and produced by that date. To provide the required data for the system analysis, conceptual designs based on the expected technology must be developed. It may be expected that large scale system studies would be used to identify the need for new wheeled amphibian designs and define their performance requirements.

d. Technological Projections. The technological projection is intended to determine the probable affects of technological advances on the characteristics of wheeled amphibians. A simple example of this might be the determination of the effect on the characteristics of a 5-ton payload amphibian if the design were based on the use of a new engine with advanced characteristics. This type question is best answered with a conceptual design which makes use of the proposed technological advance. The technological projection study may provide impetus for the development of new components or designs based on new components if it indicates large gains in effectiveness are possible.

Depending on the purpose, a large number of conceptual designs may be necessary to develop the needed information and the conceptual design procedure must be such that it can be carried out rapidly with minimum effort. In many cases the conceptual design procedures are programmed for a computer.

5-1.2 RELATION TO DESIGN CYCLES

The parametric studies or conceptual design form the first phase of the design development process and serve the purposes described in par. 5-1.1. In a design which will continue into the preliminary and detail phases, the conceptual design phase must provide information on expected technical characteristics, performance, and any basic trade-off studies performed. The conceptual design phase should also identify critical problem areas which will require modification during the following design phases. In this connection it is important to note that design assumptions made during the conceptual design should be conservative in nature. It has been shown repeatedly that a design that is marginal in meeting performance objectives at this early phase will not meet them in the later phases.

5-1.3 DESIGN CRITERIA

Many factors must be considered in the selection and design of the amphibian configuration and its systems, subsystems, and components. The relative importance of each factor will depend on the particular requirements and goals, i.e., criteria, that have been prescribed for a given concept. Factors to be considered include, but are not limited to, simplicity, durability, vulnerability, weight, arrangements, safety (human factors), maintainability, reliability, availability, effectiveness, and cost. These factors are not independent. For example, ease-of-maintenance, reliability, and economy are all associated with simplicity.

Criteria are most useful in trade-off studies or evaluation when the factors can be compared on a quantitative basis, i.e., a numeric value may be assigned to the variables. In many cases it is impossible to find a quantitative measure so that comparison must be made on a strictly qualitative basis. The paragraphs which follow define the various factors which are used as design criteria and indicate where quantitative evaluations may be made.

5-1.3.1 Simplicity

Simplicity of design leads to reliability and economy, and eases the problems of operation and maintenance. The desire for improved performance is always at odds with simplicity because these improvements inevitably arise from increased sophistication and complexity. The trade-off between simplicity and performance is quite difficult because simplicity is generally a qualitative factor. However, every effort should be made to limit complexity to a minimum consistent with the attainment of desired performance and cost objectives.

5-1.3.2 Durability

Durability is associated with the ability of the amphibian (and its components) to operate under adverse conditions over an extended period of time and yet remain free from material failures, i.e., its life expectancy. Unfortunately, this attribute usually evolves from additional material weight.

The most useful quantitative measure of durability is the statistical mean-time-between-failures (MTBF). The MTBF of a component is found by testing statistical samples of the component. The "confidence" level ascribed to the value of MTBF is a function of the number of sample components which are tested to failure. The greater the number of samples, the higher the level of confidence. Experience with similar components can be used to predict the MTBF in lieu of tests in which case the confidence level depends on the degree of experience with similar components and their similarity to the new component.

5-1.3.3 Vulnerability

Components which are susceptible to damage should be afforded any protection which does not unduly compromise normal operation. For example, sensitive electronic equipment may be specially mounted to isolate it from vibration damage. Trade-offs must be studied in some cases, e.g., housing the propeller in a tunnel certainly decreases its vulnerability, but the question arises as to what extent are propulsive efficiency and maneuvering characteristics degraded by this arrangement.

5-1.3.4 Weight

An amphibian is heavier than either a truck or boat with comparable payload capacity. The

wheeled amphibian must carry wheels and suspension systems not found on boats; a hull and waterborne propulsion system are not necessary on trucks. As a result, weight is a critical factor in the selection of every component. Of course, quantitative comparisons are easily made with respect to weight.

5-1.3.5 Arrangements

The multiplicity of systems on an amphibian require that careful consideration be given to arrangement of components and systems.

Qualitative considerations include: placement of components with easy access for maintenance; location of operator's station to afford a good field of view; location of cargo deck (and ramp) for easy loading; and keeping all parts of the electrical system out of the bilges where they would be subjected to corrosive salt water environment.

Quantitative considerations are the amphibian stability and trim which are functions of the weight and center of gravity of all the components taken collectively. Strict limitations are always set on the stability of an amphibian.

5-1.3.6 Safety (Human Factors)

There are many aspects of the amphibian design which must take into consideration the safety and comfort of the operators and support personnel. The main hazards are capsizing in surf and the physiological effects of intensive accelerations, vibrations, and noise. For example, the vibration criteria for Army land vehicles is that the allowable energy level absorbed by personnel be limited to 6 watts. Other human factors are discussed in Refs. 2 and 3.

5-1.3.7 Maintainability

The environment in which the amphibian operates decreases the life expectancy of components and increases maintenance and replacement of parts. Surf operations, in particular, with frequent groundings on beaches and coral reefs give rise to short life expectancies for unprotected propellers. The replacement of a screw on an amphibian is ordinarily quite routine. However, the LARC XV was designed with the bearing and support struts aft of the propeller so that the shaft must be uncoupled and drawn from the stuffing box to make the propeller change. Any item under consideration for future amphibian designs should be rated according to its ease-of-maintenance and its contribution to the total amphibian maintenance requirement.

A quantitative measure of maintainability is the mean-time-to-repair (MTTR), another statistical concept.

5-1.3.8 Reliability

Reliability is the probability that an item will successfully perform its intended function for a specified time interval (mission time) under stated conditions. Logistic amphibians are normally required to operate for two 10-hr mission periods, each 24-hr day, without a failure that will interrupt the mission.

Reliability is a function of durability and mission length. The reliability R of an item with an exponential distribution of failures is given by:

$$R = e^{-t/m} \tag{5-1}$$

where

t = mission time, hr
m = MTBF, hr

Thus, the reliability of an amphibian having 150-hr MTBF and 10-hr mission time is:

$$R = e^{-10/150} = 0.936 \text{ or } 94\%$$

Conversely, the expected number of failures is six per one hundred missions.

The reliability of an amphibian is a complex function of the reliability of each component and system. Refs. 4 and 5 give detailed discussion of multi-component reliability.

Eq. 5-1 is based on the simplifying assumption that the failure rate has an exponential distribution so that the probability of failure in a time interval of duration Δt is unique, i.e., dependent on the length of the interval only; independent of the starting time of the interval. This is a poor assumption for many items that show signs of wearing out as revealed by sharp increases in the probability of failure with increasing time. For example,

rubber tires have a much greater probability of blow-out after the tread has worn down to the cords. Periodic replacement of such items is necessary to maintain the desired level of reliability.

5-1.3.9 Availability

Availability (operational) is the probability that a system or item when used under stated conditions and in an actual supply environment shall operate satisfactorily at any given time. Availability (operational) A_o is a function of durability, maintainability, and the procurement time of unavailable parts:

$$A_o = \frac{m}{m + m_d} \tag{5-2}$$

where

m = MTBF, hr

m_d = mean downtime (MDT) including supply downtime, administrative downtime, and mean-time-to-repair; but preventive maintenance downtime is zero; hr

MDT is given by:

$$m_d = k\left[m_r + (1 - A_p)m_s\right], \text{ hr} \tag{5-3}$$

where

k = ratio of normal operating hours to total hours, nd

m_r = MTTR, hr

A_p = average parts availability, nd

$1-A_p$ = average parts unavailability, nd

m_s = mean procurement time for repair parts, hr

For example, the availability of an amphibian having 150-hr MTBF required to operate 20 hr each 24-hr day; and assuming MTTR = 3 hr, A_p = 0.85, and m_s = 20 days or 480 hr:

$$A_o = \frac{150}{150 + \frac{20}{24}[3 + (0.15)480]} = 0.71 \text{ or } 71\%$$

In this example, the main factor limiting the availability is the supply delay time, $k(1-A_p)m_s$.

5-1.3.10 Effectiveness

The effectiveness of a logistic amphibian is associated with its productivity. The effectiveness is measured in terms of transported cargo in ton-miles. This is the product of payload, speed, and time in operation. The speed should be the average speed for a complete cycle including time for loading and unloading. Effectiveness may also include the corrections for reliability and availability.

5-1.3.11 Economy

Cost is probably the most important criterion in the design of an amphibian and also the most difficult to assess. For any given item or system having comparable capabilities, the economic analysis consists of the comparative costs of development, production, operation, and maintenance amortized over the life of the vehicle. If the capabilities of the system are not comparable, it may be difficult if not impossible to fix a basis for economic comparison. If one system does not meet the required functional criteria, it is totally unacceptable regardless of how economical it might be; this would be a false economy. Amphibians with different capabilities can be compared on the basis of a cost-effectiveness ratio, the cost divided by the effectiveness in dollars per ton-mile of payload transported.

5-2 CONCEPTUAL DESIGN PROCEDURES

Before starting on any phase of the design process the designer must have a clear understanding of the input information, the required output, and the procedures or steps which lead from the input to the output. All phases of the design of a wheeled amphibian are similar in that the procedure involves an iterative series of calculations, drawings, and decisions which should result in a satisfactory solution. The design phases are distinguished from one another by the degree of detail involved in the design calculations, drawings, and decisions. The conceptual phase is the most general.

5-2.1 TYPICAL DESIGN INPUTS AND OUTPUTS

The inputs to the conceptual design phase can be considered as functional inputs, goals, and

constraints. These should be given in the form of military or performance requirements such as a preliminary QMR or SDR. The functional inputs specifically define the productive requirements of a wheeled amphibian. Basically this involves a definition of payload, speed, and range. Information should be provided on the weight; volume; deck area; center of gravity and character of the payload; and the performance requirement in terms of waterborne and land speed, slope and grade climbing ability, soft-soil mobility, surfing ability, and range or endurance.

The design goals in the conceptual phase are general statements of desired characteristics which are to be considered in designing to meet the functional inputs. These goals, for example, involve the objective of obtaining the vehicle of minimum size, weight, and cost; and maximum reliability and maintainability that will satisfy the functional requirements.

The design constraints are limitations which have, for some reason, been placed on the technical characteristics of the design. These might include such things as limitations on the beam, weight, clearances, engine, and materials. These constraints are almost always imposed because of considerations of transportability, mobility, or compatibility with other equipment. It must be realized that when constraints of this nature are imposed on the design, it may not be possible to meet all of the functional requirements. Thus, one of the important results of the conceptual design phase is the identification of the conflicts between the design constraints and the functional requirements.

The outputs of the conceptual design of a wheeled amphibian include a summary of the expected design characteristics and performance, simple arrangement sketches, and all supporting data and calculations.

5-2.2 BASIC DESIGN ITERATION CYCLE

The basic design iteration cycle involves the systematic variation of the design characteristics until the functional requirements and constraints are satisfied. The variation of the design characteristics in the iteration is based on the following fundamental conditions which must exist for a wheeled amphibian: (1) weight must equal buoyancy, (2) the volume and area of all the components must be equal to or less than the available volume in the hull, (3) the center of gravity and buoyancy must lie on the same vertical line, and (4) the waterborne and side slope stability must be satisfactory.

The basic steps in the conceptual design process and the iteration loops are shown in the flow chart presented in Fig. 5-1. With the input data available, the first step will be to estimate the length, breadth, and depth of the hull and select the hull form type. These first estimates are based on experience with past designs with similar payloads. The second step will be to estimate the gross vehicle weight and center of gravity location. These estimates may be based on the payload-gross weight ratios of previous wheeled amphibians. These initial steps serve to establish an initial condition from which the design iteration can proceed.

The first step in the iteration cycle is the selection of the running gear arrangement and components, based on the estimate of dimensions, weight, and centers. This selection includes the number of wheels, their location, wheelbase, and tire size. The mobility requirements in the functional inputs will provide the criteria for the land gear arrangement. The next step in the cycle will be to estimate the power required to meet the performance inputs. In general, the power required to meet the waterborne speed will govern. In some cases a particular engine will be specified so that an estimate of the expected performance will be required.

With the information available at this stage, a calculation of the gross vehicle weight and center of gravity can be made. This calculation must be checked against the gross vehicle weight and center of gravity assumed in the selection of land gear and power required. If the two do not agree within a few percent, the design loop must be repeated until agreement is reached.

After the completion of the inner iteration loop, a calculation of the waterborne stability and trim can follow. This calculation is of great importance since the stability and trim criteria must be satisfied. In general, the stability and trim calculated initially will not agree with the criteria. As a result, some modifications in the dimensions and arrangements will be necessary. At this point the volume and area required by all components should be checked against the available volume and area in the hull. A simple

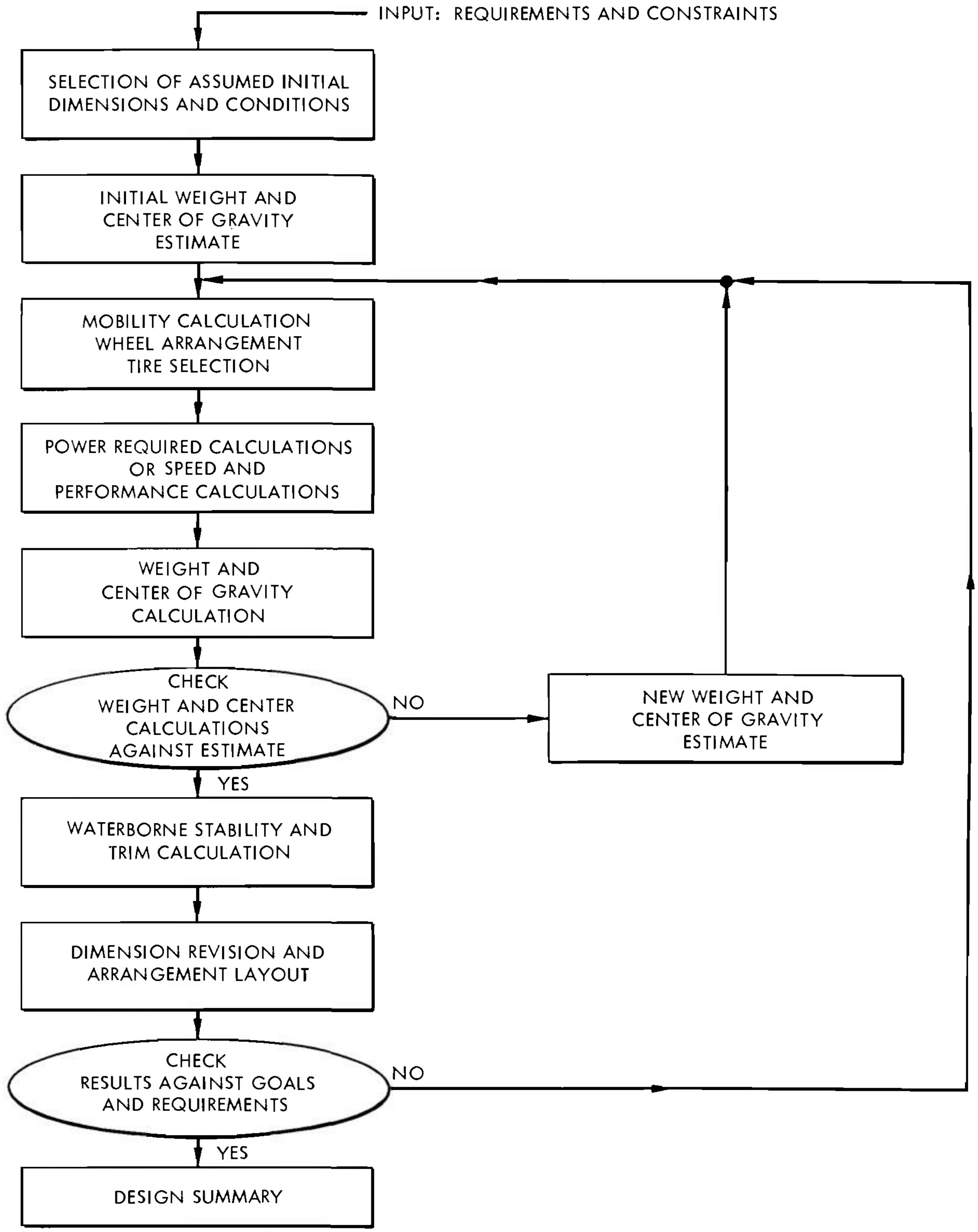


Figure 5-1. Conceptual Design Flow Chart for Wheeled Amphibians

arrangement sketch can be made for this purpose. If it is necessary to revise the dimensions or arrangements, the iteration must be repeated from the beginning point. Of course with each time through the cycle, the information from preceding cycles is available to aid in making the required calculations, selections, and decisions. In this way, the process is made to converge to a design which satisfies the functional inputs, goals, and constraints. When this point is reached, a design summary is prepared.

The remaining paragraphs of this chapter present some of the data and calculation procedures needed to carry out the steps in the conceptual design procedure discussed.

5-3 SIZE AND ARRANGEMENTS

In general, the data and calculation procedures used in the conceptual design procedure are developed from experience with past successful designs in order to develop the conceptual design output in the shortest time with a minimum of manpower. For example, during preparation of the conceptual design, hull structural plans will normally not be developed to the state that a weight estimate can be prepared by direct calculation. Instead, the hull weight estimate will be made from known relationships among size, shape, and weight variations developed from data on past designs. As a result, conceptual design data must be updated each time information on a new design becomes available. Further, as the state-of-the-art with respect to vehicle components advances, the conceptual design data must be updated. This applies to such items as engines, transmission systems, running gear, and controls.

The designer must be careful that the use of data from past vehicles, in the designing of new vehicles, does not inhibit his use of new concepts and developments. In order to take advantage of such advances, it may be necessary to carry out some steps of the conceptual design in much greater detail than normally required. For example, when studying the effect of Fiberglas hull construction on vehicle characteristics, data on the weights of existing steel and aluminum hulls would not be of much use. It would be necessary to lay out a typical Fiberglas structure in sufficient detail to calculate a realistic weight. However, it should not be necessary to develop special data on items such as engines, drive trains, and running gear.

5-3.1 DESIGN INTERRELATIONSHIPS

Before undertaking a conceptual design, the designer must have a thorough understanding of the basic interactions between the performance and design characteristics. This knowledge is required for making the selections and compromises to produce a rapid convergence of the iterative parts of the design cycle. A summary of these interactions is presented in Ref. 6 for high water speed, planing hull amphibians. These interactions have been modified to apply to conventional wheeled amphibians and are presented in Figs. 5-2 through 5-7.

Fig. 5-2 presents the interactions between design characteristics and performance in terms of primary or secondary influence of a given design characteristic on the indicated performance characteristic. Fig. 5-3 presents a more specific indication of the interrelation of hull design characteristics and performance. This figure is arranged to indicate the effects of a change in a hull design feature on a given performance characteristic. Of course, various hull design features are not independent of one another. Fig. 5-4 presents the interrelationships among hull design features in terms of primary or secondary influences.

The running gear will affect both land and waterborne performance. Fig. 5-5 shows the effect of an increase or change in running gear characteristics on performance. Fig. 5-6 presents the interrelationships among running gear features in terms of primary or secondary influences. The final figure dealing with design interrelations is Fig. 5-7 which indicates the primary and secondary influences of mechanical features.

5-3.2 AREA, VOLUME, AND ARRANGEMENT RELATIONSHIPS

During conduct of the conceptual design it is necessary to determine the area and volume required for the major components and to

DEPENDENT \ INDEPENDENT	WATER FEATURES					LAND FEATURES				CARGO FEATURES		
	GROSS VEHICLE WEIGHT	HULL		MARINE POWER SYSTEM	MARINE PROPULSION SYSTEM	RUNNING GEAR		LAND POWER SYSTEM	LAND STEERING SYSTEM	AREA	WET DECK	RAMP
		SIZE	FORM			SIZE	CONFIGURATION					
LOAD CAPABILITY	1	1	2	2		2	2	2		1	2	1
WATER PERFORMANCE	1	1	1	1	1	2	2				1	2
SURF & BEACHING PERFORMANCE	1	2	2		1	1	1		2		1	2
OFF-ROAD PERFORMANCE	1	2	2			1	1	2	2		2	2
ON-ROAD PERFORMANCE	2	1				2	2	2	2			
TRANSPORTABILITY	1	1				2						

1 - PRIMARY INFLUENCE
2 - SECONDARY INFLUENCE

Figure 5-2. Design Characteristic and Performance Interaction

arrange these in a logical fashion. These major components are the payload, machinery, running gear, crew, and marine propulsor.

The payload or cargo that a wheeled amphibian can carry is determined by its weight, vertical center of gravity, and the required deck area. In many cases both the cargo weight and its area will be given as input to the conceptual design. In the event that only the weight of cargo is given, Fig. 5-8 can be used to estimate the area required. This figure presents the design weight of payload per square foot of cargo deck area for the production wheeled amphibians. Data are also given on the unit weight of actual cargo carried by the LARC V and LARC LX in Southeast Asia in late 1967 and early 1968. In general, the cargoes have been lighter than the vehicles were designed to carry, i.e., load limitations are deck area rather than weight. Data are also presented in Fig. 5-8 on the loadings of CONEX and A.S.A. standard 8 X 8 X 20 ft commercial containers.

In general, the block volume and planform area of typical machinery items such as engines, transmissions, and transfer cases are readily available from manufacturers' data. Typically the power per cubic foot of block volume (block volume defined as overall length X overall width X overall height) of current production Diesel engines range from 4 to 8 BHP per cubic foot and 8 to 10 BHP per cubic foot for current production gas turbine engines. The power per square foot of planform area (planform area defined as overall length X overall width) of current production Diesel engines ranges from 14 to 20 BHP per square foot and 30 to 50 BHP per square foot for current production gas turbines.

During the conceptual design it is necessary to estimate the area and volume of the machinery space. This requires data such as that in Table 5-1 which presents the ratios of block machinery volume and planform area, and the machinery compartment volumes and areas for several wheeled amphibians. Data are also presented in Table 5-1 on the area and volume of the radiator compartments for the wheeled amphibians with liquid-cooled engines.

In a wheeled amphibian the crew is stationed in a cab which provides protection from the

HULL DESIGN VERSUS PERFORMANCE	WATER PERFORMANCE				LAND PERFORMANCE											
+ = BETTER	RESISTANCE	SEA KINDLINESS	STABILITY	SURF ABILITY AND BEACHING	SOFT SOIL MOBILITY	GROUND CLEARANCE	APPROACH ANGLE	BREAK ANGLE	DEPARTURE ANGLE	RIDE	ROADABILITY	SIDE SLOPE STABILITY	LONGITUDINAL STABILITY	AGILITY	LOAD PERFORMANCE	TRANSPORTABILITY
LENGTH	+	+	+	+			-	-	-	+	-		+	-	+	-
BEAM (OVERALL)	-	-	+	+						+	-	+		-	+	-
GROSS VEHICLE WEIGHT	-	+	-	+	-					+	-	-		-	+	-
LONGITUDINAL CG FORWARD OF STERN							+		-							
BOW SHAPE, FULLNESS	-	-	+	+			-							-		
STERN SHAPE, FULLNESS	-		+	+					-					-	+	
FREE BOARD			+	+											-	-
VCG, HEIGHT			-	-								-			-	

Figure 5-3. Interrelation of Hull Design and Performance

DEPENDENT \ INDEPENDENT	LENGTH	BEAM	APPROACH & DEPARTURE ANGLE	GROSS VEHICLE WEIGHT *	LCG	BOW SHAPE	STERN SHAPE	FREE BOARD	VCG	RUNNING GEAR	WATERBORNE STABILITY	RAMP
LENGTH		1		1		2	2			2	1	2
BEAM	1			1		2	2		1	1	1	2
APPROACH & DEPARTURE ANGLE						1	1			2		
GROSS VEHICLE WEIGHT *	1	1	2			2	2	1	2	1	1	2
LCG	2									1		
BOW SHAPE	2	2	2		2			2		2	2	1
STERN SHAPE	2	2			2			2		2	2	1
FREE BOARD	2	2	2	2					2			2
VCG	2	1	2	2		2	2	2		2	1	
RUNNING GEAR	1	1	2	1	1	2	2				2	
WATERBORNE STABILITY	1	1		1		2	2		1	2		
RAMP	2	2		2		1	1	2				

* AS AFFECTED BY HULL DESIGN ONLY
1 - PRIMARY INFLUENCE
2 - SECONDARY INFLUENCE

Figure 5-4. Interrelation of Hull Features

elements when in the water, in surf, and on land. Table 5-2 presents the planform area and volume of the cabs on several amphibians as a function of the number of seats in the cab. It should be noted that the LARC XV has different forward directions on land and water and, as a result, the cab is arranged for the driver to face in either direction, as required.

Information on the area, volume, and arrangement of the running gear and marine propulsor is given in pars. 5-5.3 and 5-5.4.

The arrangement of the major items in the hull is governed by considerations of center of gravity location, access, maintenance, and visibility. The longitudinal location of the center of gravity of the loaded amphibian should be such that it floats with no trim or a little trim by the stern. The running gear should be located so that the loading on the wheels is equal. The vertical center of gravity should be as low as possible consistent with other requirements such as freeboard.

INDEPENDENT / DEPENDENT (+ = BETTER)		LAND PERFORMANCE										WATER PERFORMANCE				
		SOFT SOIL MOBILITY	GROUND CLEARANCE	APPROACH ANGLE	BREAK ANGLE	DEPARTURE ANGLE	RIDE	ROADABILITY	SIDE SLOPE	GRADE	AGILITY	SURF AND BEACHING	SPEED	SEA KINDLINESS	STATIC STABILITY	LOAD PERFORMANCE
NUMBER OF TIRES (SINGLES)		+	-		+		+	+	+	+	+		+		+	
TIRE	DIAMETER	+	+	+	+	+	+		-			+	-		-	-
	SECTION WIDTH	+					+		-			+	-		-	
SUSPENSION DEPTH		+					+		-	-					-	-
STEERING MODE	ACKERMANN							+			+					
	ARTICULATED FRAME	+													+	
WHEEL DRIVE	DIRECT	+														
	FRICTION						-								+	+
GROSS VEHICLE WEIGHT *		-									-	-	-		-	-

*AS AFFECTED BY RUNNING GEAR DESIGN

Figure 5-5. Running Gear Changes vs Performance

INDEPENDENT / DEPENDENT	GROSS VEHICLE WEIGHT *	NUMBER OF TIRES	TIRE DIAMETER	TIRE SECTION WIDTH	WHEEL DRIVE ARRANGEMENT	STEERING MODE	SUSPENSION DEPTH	HULL
GROSS VEHICLE WEIGHT *		2	2	2	2	1	1	
NUMBER OF TIRES	1		1	1	2	2	2	1
TIRE DIAMETER	1	1		1	2	1	2	1
TIRE SECTION WIDTH	1	1	1			2		1
WHEEL DRIVE ARRANGEMENT	2	1	2			1	1	2
STEERING MODE	2	2	1					1
SUSPENSION DEPTH		2	2	2	1	2		
HULL		1	1	1	2	1		

* AS AFFECTED BY RUNNING GEAR DESIGN
1 - PRIMARY
2 - SECONDARY

Figure 5-6. Running Gear Features

INDEPENDENT / DEPENDENT	GROSS VEHICLE WEIGHT	HULL SIZE	HULL FORM	RUNNING GEAR SIZE	RUNNING GEAR CONFIGURATION	POWER SYSTEM, MARINE	POWER SYSTEM, LAND	WATER PROPULSION CONFIGURATION	LAND STEERING CONFIGURATION
GROSS VEHICLE WEIGHT		1		1	2	2	2	2	2
HULL SIZE	1			1	2	2		2	2
HULL FORM				1	1	2		2	2
RUNNING GEAR SIZE	1	1	2		1			2	1
RUNNING GEAR CONFIGURATION	2	2	1	1			2		2
POWER SYSTEM, MARINE	1	1	1	2				1	
POWER SYSTEM, LAND	1			2	2				2
WATER PROPULSION CONFIGURATION	2	2	2	2	2	1			
LAND STEERING CONFIGURATION			2	1	1		2		

1 - PRIMARY INFLUENCE
2 - SECONDARY INFLUENCE

Figure 5-7. Interrelation of Mechanical Features

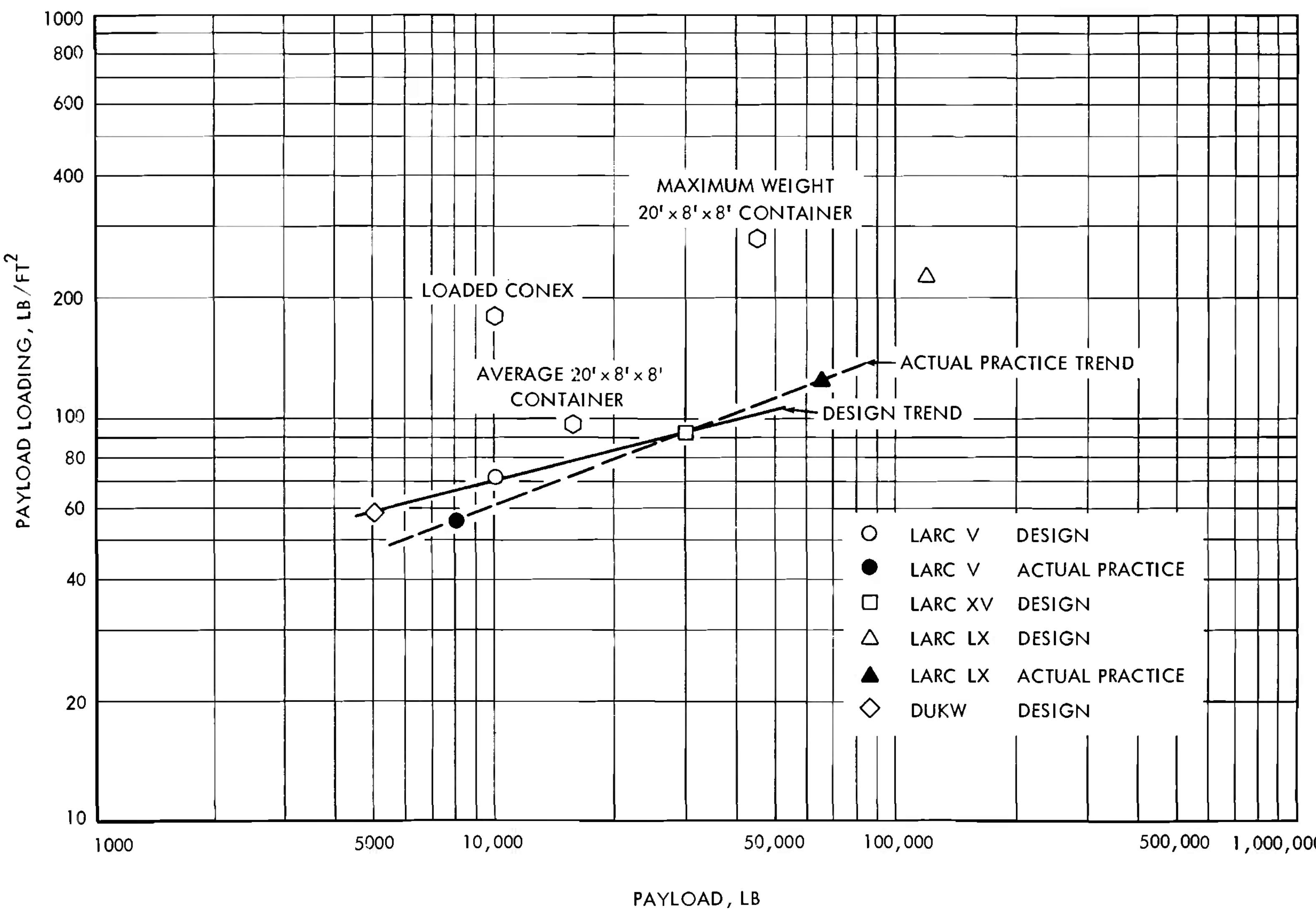


Figure 5-8. Wheeled Amphibian Cargo Loading

TABLE 5-1 ENGINE COMPARTMENT VOLUMES AND AREAS

Vehicle	*Engine Type*	*Max BHP*	*Cooling*	$\frac{A_{eng}}{A_{comp}}$	$\frac{V_{eng}}{V_{comp}}$	$\frac{A_{cool}}{BHP}$, ft^2/BHP	$\frac{V_{cool}}{BHP}$, ft^3/BHP
LARC V	Diesel	300	Liquid	0.33	0.264	0.058	0.172
LARC XV	Twin Diesel	600	Liquid	0.308	0.204	0.060	0.248
LARC LX	Four Diesels	660	Liquid	0.224	0.128	0.127	0.508
LVH-X2	Gas Turbine	1100	Air	0.47	0.260	—	—

A_{eng} = planform area of machinery in compartment, ft^2

A_{comp} = planform area of compartment, ft^2

V_{eng} = block volume of machinery in compartment, ft^3

V_{comp} = volume of compartment, ft^3

A_{cool} = planform area of radiator and ducting, ft^2

V_{cool} = volume of radiator and ducting, ft^3

TABLE 5-2 CREW CAB DATA FOR WHEELED AMPHIBIANS

Vehicle	*Seats in Cab*	$\frac{\text{Cab Area}}{\text{Seat}}$, ft^2	$\frac{\text{Cab Vol}}{\text{Seat}}$, ft^3
LARC V	3	11.3	62.7
LARC XV	4	14.9	70.8
LARC LX	1	28.0	140.0
LVH-X2	2	16.8	95.0

The usefulness of an amphibian is strongly affected by the ease with which it can be maintained and repaired. Good maintainability is assured by providing sufficient access and working space around those items which require maintenance. This implies that hatches should be located over major components such as engines and transmissions. In general, locating the cab over major components will reduce access to such components and complicate the running of control systems. The cab should be located so that the driver's visibility is not obstructed by the cargo or the amphibian structure. The number of components located under hatches on which cargo will be loaded should be minimized. Such hatches are often damaged and leak.

5-3.3 DIMENSION AND COEFFICIENT RELATIONSHIPS

The hull dimensions are a function of the volume, area, and arrangement requirements of the major components, as covered in par. 5-3.2, as well as the required buoyancy, stability, and trim. Any one of these items could have the effect of fixing a minimum value for one of the principal hull dimensions which include the

overall length, the waterline length, the overall breadth, the waterline breadth, the hull depth at various locations, and the draft to the hull bottom at various locations.

5-3.3.1 Buoyancy

When an amphibian is floating in the static equilibrium, Archimedes' Law requires that its weight be equal to the weight of the displaced fluid, i.e., buoyancy. The weight of the displaced fluid is directly proportional to the volume of the amphibian below the free surface, which, in turn is related to the dimensions. During the conceptual design phase, the dimensions needed to obtain the required buoyancy are estimated without preparing detailed drawings of the hull.

In general, the buoyancy for different types of amphibian hulls can be estimated by using the "block coefficient" which is representative of the particular hull type. The block coefficient C_B is defined by:

$$C_B = \frac{GVW}{LBH_m \rho g}, \text{ nd} \tag{5-4}$$

where

GVW = gross vehicle weight, lb
ρg = density of fluid, lb/ft^3
L = overall or waterline length, ft
B = overall or waterline breadth, ft
H_m = mean draft, ft

The value of the block coefficient used must be consistent with the choice of overall or waterline dimensions. In general, it is preferable to use the waterline dimensions if available since this avoids problems with adjustments for overhangs and bumpers. Table 5-3 contains information on the dimensions and block coefficients of existing wheeled amphibians. These data can be used for estimates of reasonable block coefficients for new designs. It should be noted that the block coefficient of the DUKW is lower than the LARC's despite its scow-type hull form. This is the result of the proportionately larger wheel wells and tunnels required by the 6 X 6 arrangement of the running gear and the sprung suspension.

In order to estimate the hull depth at different locations, it is necessary to determine the required freeboard as well as mean draft. The freeboard requirements are a function of the range of stability and reserve buoyancy criteria which are presented in par. 7-1.6.2. At this stage in the conceptual design there will be insufficient information to calculate the range of stability and reserve buoyancy to check an assumed freeboard. Accordingly, it is necessary to base an estimate of the freeboard on data from past designs, as given in Table 5-4.

It should be noted that the distribution of freeboard is a function of the arrangement of the cargo deck. Two arrangements of cargo decks have been used for wheeled amphibians, viz., the "wet deck" and the cargo compartment arrangements. In the wet deck type, the cargo is carried on a watertight deck located above the load water line. Water on deck from surf or waves will run overboard without entering the bilges. The LARC V and LARC XV have this type of arrangement.

In the cargo compartment arrangement, the cargo is carried on a deck below the waterline and is protected by high freeboard and coamings. Water which enters the cargo compartment from surf and waves flows into the bilge and must be pumped out. The LARC LX and the DUKW are arranged in this manner.

The wet deck arrangement has the advantage of being much safer in surf or high seas, and unloading can take place without the necessity for lifting the cargo over the high sides and coamings. This arrangement has the disadvantage of placing the center of gravity of the cargo relatively high, which requires an increase in hull dimensions for stability. Conversely, the cargo compartment arrangement provides for a reduced height of the cargo center of gravity, thus reducing the hull dimensions. It has the disadvantage of the possibility of swamping in high surf or waves and relatively greater difficulty in unloading.

From Table 5-4 it may be noted that the freeboard forward is greater than the freeboard aft. This is the result of trim and not due to greater hull depth forward. In general, when estimating the freeboard forward and aft for the purpose of establishing the hull depth, it is proper to use the mean of the bow and stern freeboard ratios. It is desirable for the forward and aft freeboard to be about the same, with the forward freeboard slightly greater, so that large trim angles do not develop at large heel angles.

TABLE 5-3 CHARACTERISTICS OF PRODUCTION WHEELED AMPHIBIANS

Vehicle	Gross Vehicle weight, lb	Payload, lb	Payload / GVW, nd	Length Overall	Length Waterline	Beam Overall	Beam Waterline	Mean Hull Draft	BHP	Speed, mph	Hull Depth: Forward	Hull Depth: Midships	Hull Depth: Aft
LARC V	31000	10,000	0.323	35'-0"	32'-0"	10'-0"	9'-8"	2'-3"	300	8.[illegible]	5'-10¾"	3'-1¼"	5'-5"
LARC XV	75[illegible]	30,000	0.398	45'-0"	39'-[illegible]"	14'-6"	14'-0"	2'-8½"	600	9.7	7'-0"	3'-10"	6'-0"
LARC LX	314000	120,000	0.382	62'-5"	58'-9"	26'-11"	25'-4"	5'-1"	660	7.0	3'-6" Side 9'-9"	4'-3" 9'-9"	9'-7¼"
DUKW	20,000	5,000	0.25	31'-0"	29'-0"	8'-4"	8'-0"	2'-6"	90	6.5	3'-9"		4'-0"

Vehicle	Running Gear Arrangement	Tire Size, in.	Block Coef. Based on Overall Dim., nd	Block Coef. Based on Waterline Dim., nd	C_{I_T} * Based on Overall Dim., nd	C_{I_T} * Based on Waterline Dim., nd
LARC V	4 x 4	18.00 x 25	0.610	0.690	0.051	0.055
LARC XV	4 x 4	24.00 x 29	0.658	0.776	0.0528	0.067
LARC LX	4 x 4	36.00 x 41	0.573	0.649	0.0495	0.063
DUKW	6 x 6	11.00 x 18	0.480	0.557	0.070	0.075

* C_{I_T} = Transverse Inertia Coefficient

TABLE 5-4 FREEBOARD RATIOS, WHEELED AMPHIBIANS
(Full Load Condition)

Vehicle	*Freeboard Midships / Mean Hull Draft*		*Freeboard Forward / Freeboard Midships*	*Freeboard Aft / Freeboard Midships*
LARC V	0.39		3.42	2.58
LARC XV	0.46		3.73	2.66
LARC LX	0.935		1.05	0.95
	To Deck	*To Coaming*		
DUKW	0.61	1.00	1.21	0.79

5-3.3.2 Waterborne Stability—Intact

For a wheeled amphibian to operate safely in the water, its geometry and arrangement must be such that, when heeled, the couple produced by the buoyant force acting up and the weight acting down will tend to right the amphibian. The criteria for acceptable values of this restoring moment under different conditions, and procedures for calculating the moment, are covered in detail in pars. 7-1.6.2 and 7-1.6.3. During the conceptual design study, the time and information may not be available to perform detailed calculations and check all of the stability criteria. Therefore, the most important criteria must be selected and calculation procedures developed to check these criteria. The criterion usually considered is the initial stability expressed in terms of the *GM* or metacentric height. The concept of *GM* is covered in all textbooks on naval architecture and in pars. 7-1.6.2 and 7-1.6.3. The *GM* value is determined from the relation:

$$GM = KB + BM - KG \tag{5-5}$$

where
- KB = vertical distance from the hull bottom to the center of buoyancy, ft
- BM = vertical distance from the center of buoyancy to the metacenter, ft
- KG = vertical distance from the hull bottom to the center of gravity, ft

For wheeled amphibians a good estimate of *KB* is given by:

$$KB = 0.50\,H_m,\ \text{ft} \tag{5-6}$$

where
- H_m = mean draft of hull, ft

The *BM* is calculated from the equation

$$BM = \frac{I}{\nabla},\ \text{ft} \tag{5-7}$$

where
- I = transverse moment of inertia of the waterplane area about its centerline, ft^4
- ∇ = displaced volume of the amphibian, ft^3

Since the exact shape of the waterplane will not be known during the conceptual design, *I* may be estimated from

$$I = C_{I_T} B^3 L,\ \text{ft}^4 \tag{5-8}$$

where
- C_{I_T} = transverse inertia coefficient, nd
- B = overall or waterline beam, ft
- L = overall or waterline length, ft

Table 5-3 presents values of the transverse inertia coefficient C_{I_T} for several wheeled amphibians based on both the waterline and overall dimensions. In Table 5-3 it should be noted that the value of C_{I_T} is largest for the DUKW. This is due to the aft wheel well being completely under the free surface at the full load condition.

The vertical height KG of the center of gravity is obtained from the weight and center estimate. From this value of KG and the calculations in Eqs. 5-6, 5-7, and 5-8 the intact waterborne stability can be determined from Eq. 5-5.

5-3.3.3 Trim

Trim is defined as the difference in the drafts measured forward and aft. To float in static equilibrium, the center of buoyancy must be on the same vertical line as the center of gravity. In general, any nonsymmetrical weights can be balanced so that the center of gravity is on the longitudinal centerline. Equilibrium is thus satisfied if the longitudinal center of buoyancy and gravity, measured from the same reference point, are equal. During the conceptual design studies, values of trim will be calculated for various loading conditions.

The trim T can be calculated from the following approximate equation:

$$T = \frac{\Delta(LCB - LCG)}{MT1}\text{, in.} \qquad (5\text{-}9)$$

where

Δ = gross vehicle weight (GVW), lb

LCG = distance of center of gravity forward of the transom, ft

LCB = distance of the center of buoyancy at zero trim forward to transom, ft

$MT1$ = moment to alter trim one inch, ft-lb/in.

A positive trim calculated from Eq. 5-9 is defined as trim by the stern, i.e., draft aft is greater than draft forward.

An estimate of the value of LCB can be obtained from the data on existing wheeled amphibians given in Table 5-5.

It should be noted that the amphibian actually trims about the longitudinal center of flotation LCF, the longitudinal location of the center of area of the waterplane. The draft H at any point can be calculated from

$$H = H_m + \frac{Td}{12\,LWL}\text{, ft} \qquad (5\text{-}10)$$

where

H_m = mean draft, ft

T = trim, in.

d = distance from LCF to point that the draft is calculated, ft

LWL = load waterline length, ft

Note: If the point in question is forward of the LCF, d is (−) and if the point is aft of the LCF, d is (+).

Procedures for estimating the value of $MT1$ are presented in par. 7-1.6.4.

5-4 WEIGHTS AND CENTERS

The key step in the conceptual design process is the estimate of the vehicle weight and location of center of gravity. A realistic estimate of these items is of great importance because of their controlling effect on concept feasibility and the estimated performance of the amphibian. An unrealistic weight and center estimate will greatly increase the work required during the preliminary and detail design phases.

5-4.1 WEIGHT EQUATION AND WEIGHT GROUPS

It has been found from long experience in ship and boat design that the best way to make realistic weight and center estimates is to establish a weight equation that relates the sum of a number of weight groups to the gross

TABLE 5-5 LOCATION OF ZERO TRIM *LCB* AND *LCF* FOR WHEELED AMPHIBIANS

Vehicle	*LCB/LWL*	*LCF/LWL*
LARC V	0.495	0.485
LARC XV	0.498	0.482
LARC LX	0.490	0.498

NOTE: *LCB* and *LCF* are measured from the aft end of the load waterline.

vehicle weight. Such a weight equation for wheeled amphibians is given by

$$\text{Gross Vehicle Weight} = GVW = W_A + W_B + W_C + W_D + W_E + W_F + W_M + W_L \qquad (5\text{-}11)$$

where the subscripts refer to the major weight groups that, combined, equal the gross vehicle weight. A description of these weight groups is given in Table 5-6.

A detailed breakdown of the items included in each of the eight major groups listed in Table 5-6 is given in par. 6-6.1.

TABLE 5-6 WHEELED AMPHIBIAN WEIGHT GROUPS

Major Group	*Description*
A	Hull Structure
B	Machinery
C	Drive Train
D	Marine Propulsor
E	Running Gear and Suspension
F	Miscellaneous Systems
M	Margin
L	Loads, including fuel, crew, and payload

5-4.2 WEIGHT DATA

In order for the weight equation, Eq. 5-11, to be useful in the conceptual design, methods must be developed to estimate the total weight of each major weight group. This has been accomplished by taking "as built" weight data from existing wheeled amphibians, and separating the data into the major weight groups. These data were then related to vehicle characteristics and parameters which are known during the conceptual design stage. The resulting data are described in the paragraphs which follow.

5-4.2.1 Group A—Hull Structure

The hull weight was found to be most closely related to the parameter

$$\text{Hull Size Parameter} = \frac{LOA(BOA + D_{avg})}{100} \qquad (5\text{-}12)$$

where

LOA = length overall, ft

BOA = breadth overall, ft

D_{avg} = average depth, ft

The average depth is defined as the projected vertical profile area of the hull and cab divided by the overall length. The weights of Group A for several wheeled amphibians are plotted against this parameter in Fig. 5-9. It should be noted that the LARC LX is steel while the rest of the amphibians included in Fig. 5-9 are aluminum.

5-4.2.2 Group B—Machinery

The weight of engines varies with the type, power, date of development, and intended use. Weight data on a number of engines that could be used in a wheeled amphibian are presented in Fig. 5-10. In addition to the engine proper, other items included in Group B are listed in par. 6-6.1. For wheeled amphibians these items amount to between 1.5 and 3.0 lb/BHP for liquid-cooled internal combustion engines, and between 0.5 and 1.0 lb/BHP for air-cooled internal combustion and gas turbine engines.

5-4.2.3 Group C—Drive Train

The weight of the drive trains of several wheeled amphibians is presented in Fig. 5-11 as a function of brake horsepower. With the exception of one point, these data all apply to mechanical drive trains. The exception, which is marked, applies to the estimated weight of a hydraulic drive train proposed for the Mobile Floating Assault Bridge-Ferry in 1961.

5-4.2.4 Group D—Marine Propulsor

The weight of the items in weight Group D totals between 2.0 lb/BHP (around 300 BHP) to about 1.5 lb/BHP (around 600 BHP) for existing amphibians. These weights apply to systems using propellers in tunnels with Kort nozzles. Values in this weight group will depend greatly on the propeller diameter, rpm, length of

shafting, and type of propulsor. The use of a right angle drive system on a conventional wheeled amphibian would increase the weight of Group D to about 3.0 lb/BHP. The use of waterjets would result in a Group D weight of about 3.5 lb/BHP, including the weight of water in the pump and ducting. Of course, the weights of the right angle drive and waterjet propulsor will also depend greatly on the detail characteristics of the system.

5-4.2.5 Group E—Running Gear and Suspension System

The weights of the running gear and suspension systems of several wheeled amphibians are presented in Fig. 5-12 as a function of gross vehicle weight. The LARC series of amphibians does not have sprung suspensions but depends instead on the flexibility of the tires for spring action. The

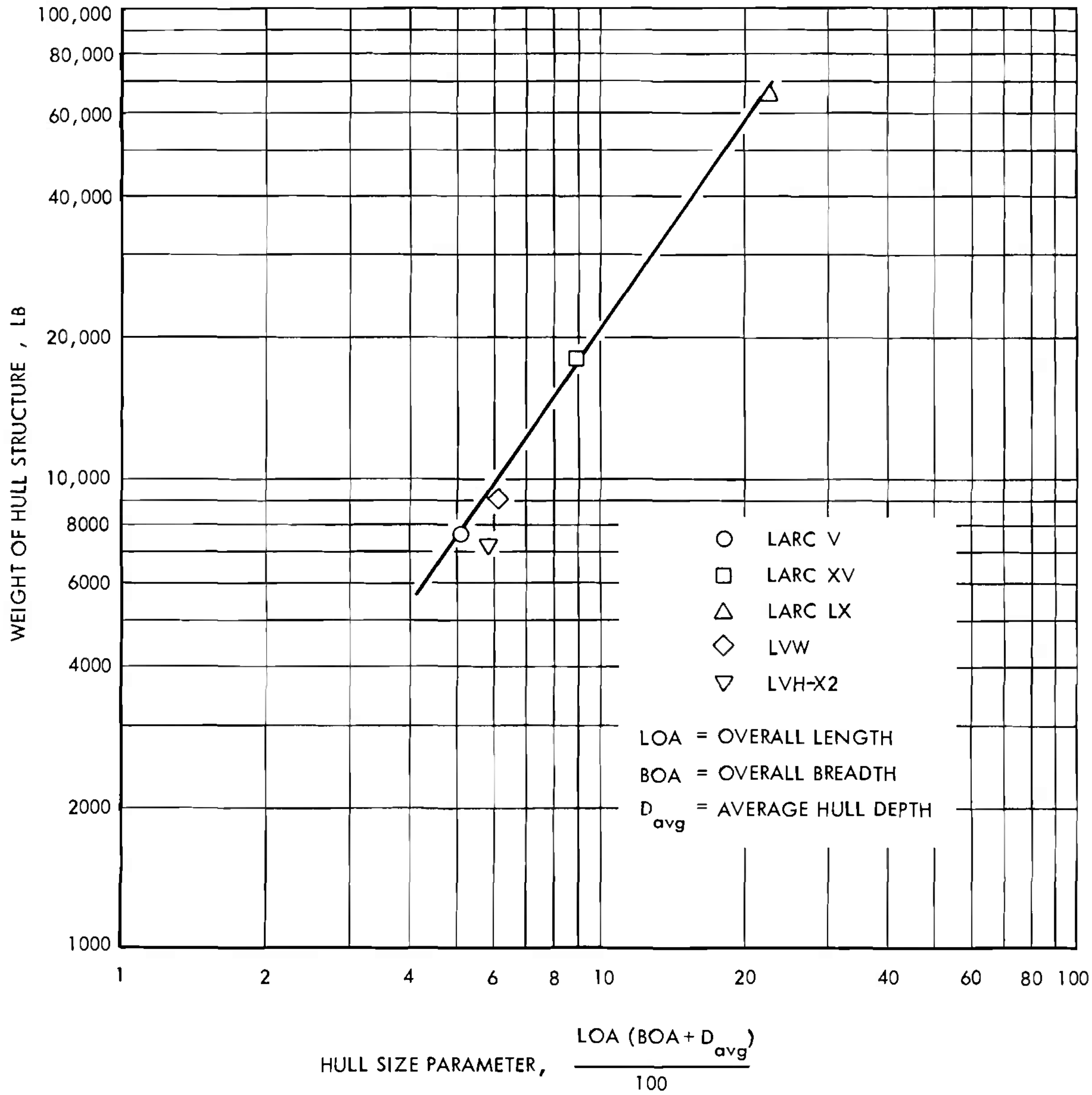


Figure 5-9. Weight Group A—Hull Structure

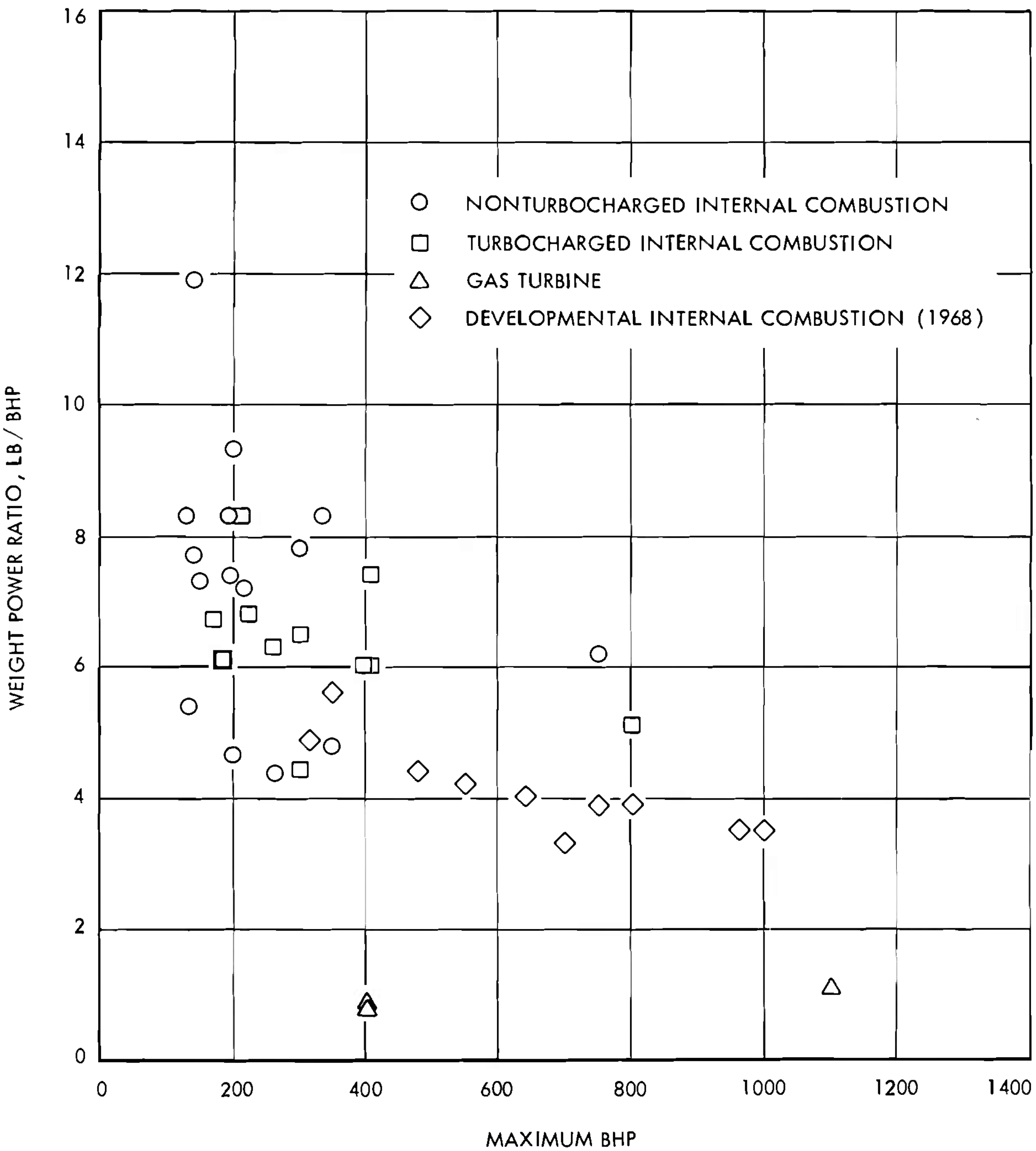


Figure 5-10. Specific Engine Weights for Multi-fuel or Diesel Engines

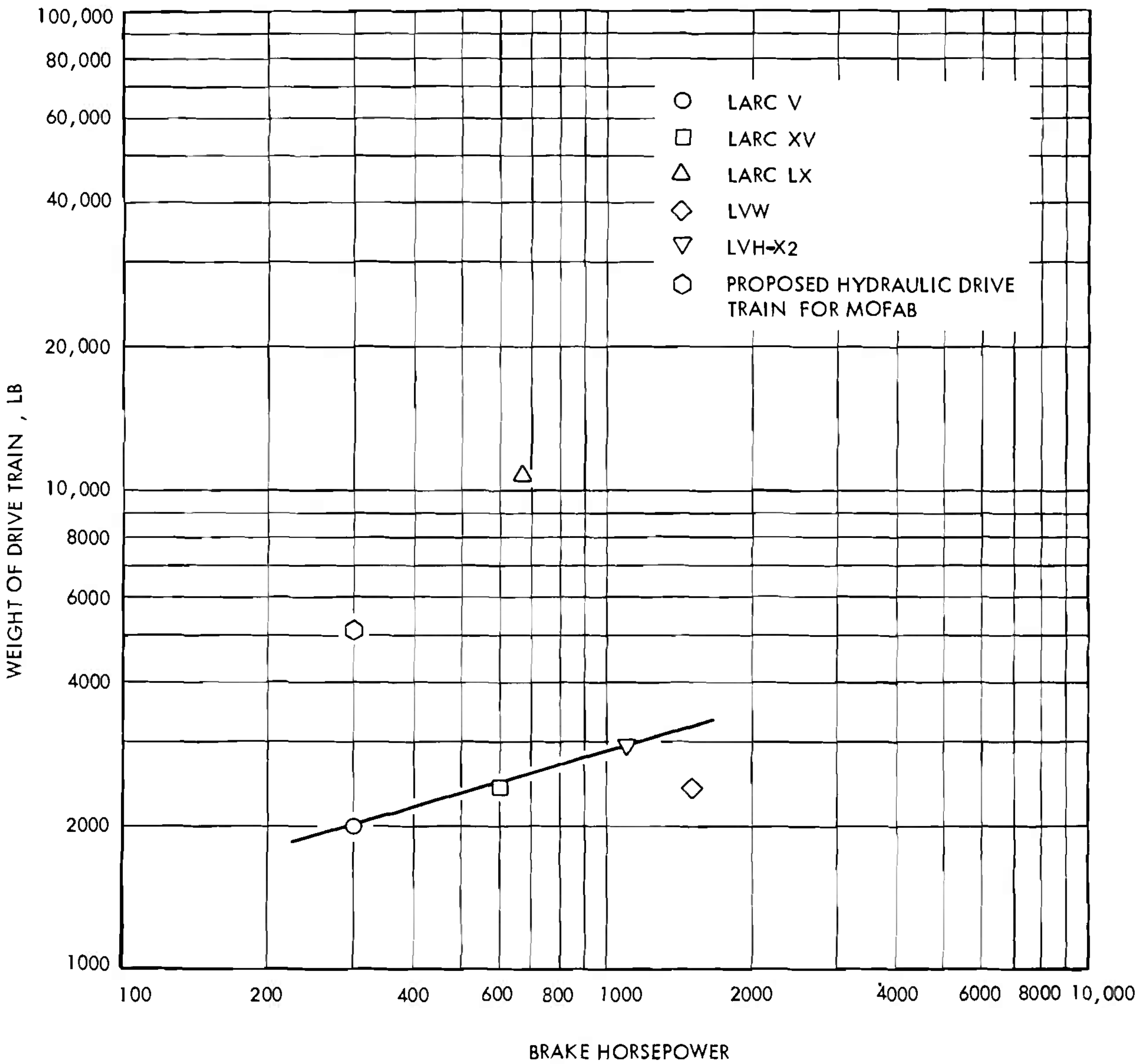


Figure 5-11. Weight Group C—Drive Train

LVW and LVH-X2 have wheel retraction systems built with very lightweight components.

5-4.2.6 Group F—Miscellaneous Systems

The weights of the miscellaneous systems are presented in Fig. 5-13 as a function of the vehicle empty weight. The LVW is equipped with a very complex hydraulic system which results in a high weight for Group F relative to the other wheeled amphibian data shown in Fig. 5-13.

5-4.2.7 Group M—Margins

The subject of weight margins is covered in par. 5-4.4.

5-4.2.8 Group L—Loads

The loads a wheeled amphibian is to carry are determined by the input requirements and machinery type. As a result loads can be calculated directly during the conceptual design studies.

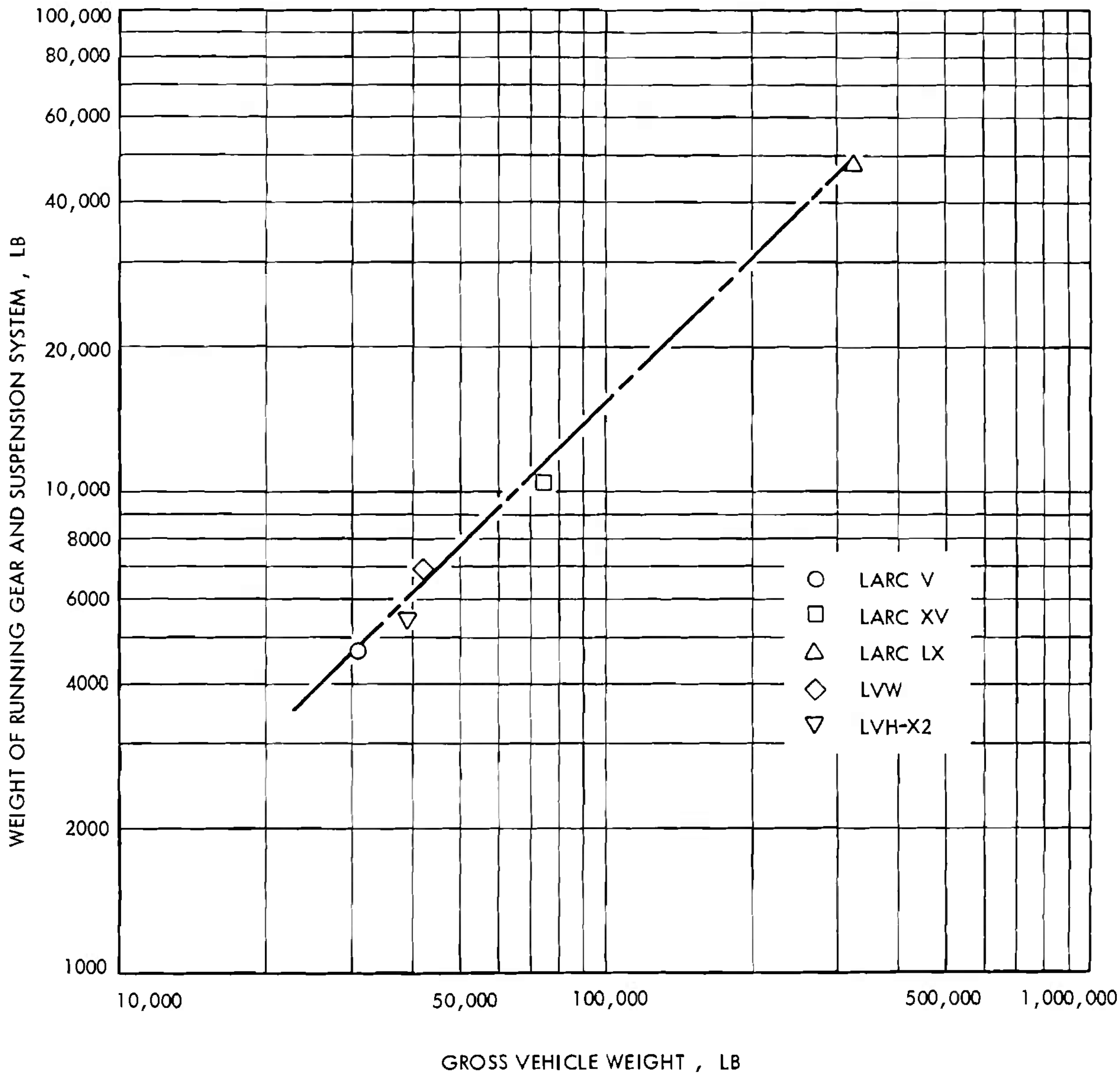


Figure 5-12. Weight Group E—Running Gear and Suspension System

5-4.3 CENTER OF GRAVITY LOCATIONS

In addition to an estimate of the gross vehicle weight, the longitudinal and vertical coordinates of the center of gravity must be determined so that the stability and trim may be calculated. This is best accomplished by estimating the center of gravity of each of the major weight groups and combining them to obtain the center of gravity of the complete amphibian.

From a simple arrangement sketch prepared during the conceptual design study, it is possible to make a good estimate of the longitudinal and vertical centers of Groups B—Machinery, C—Drive Train, D—Marine Propulsor, E—Running Gear and Suspension, and L—Loads. Data are presented in Figs. 5-14 and 5-15 on the center of gravity locations for Groups A—Hull and F—Miscellaneous Systems. In the calculation of the vertical center of gravity of the cargo, loaded CONEX containers have been assumed to have a CG height of 40 in. above the cargo deck.

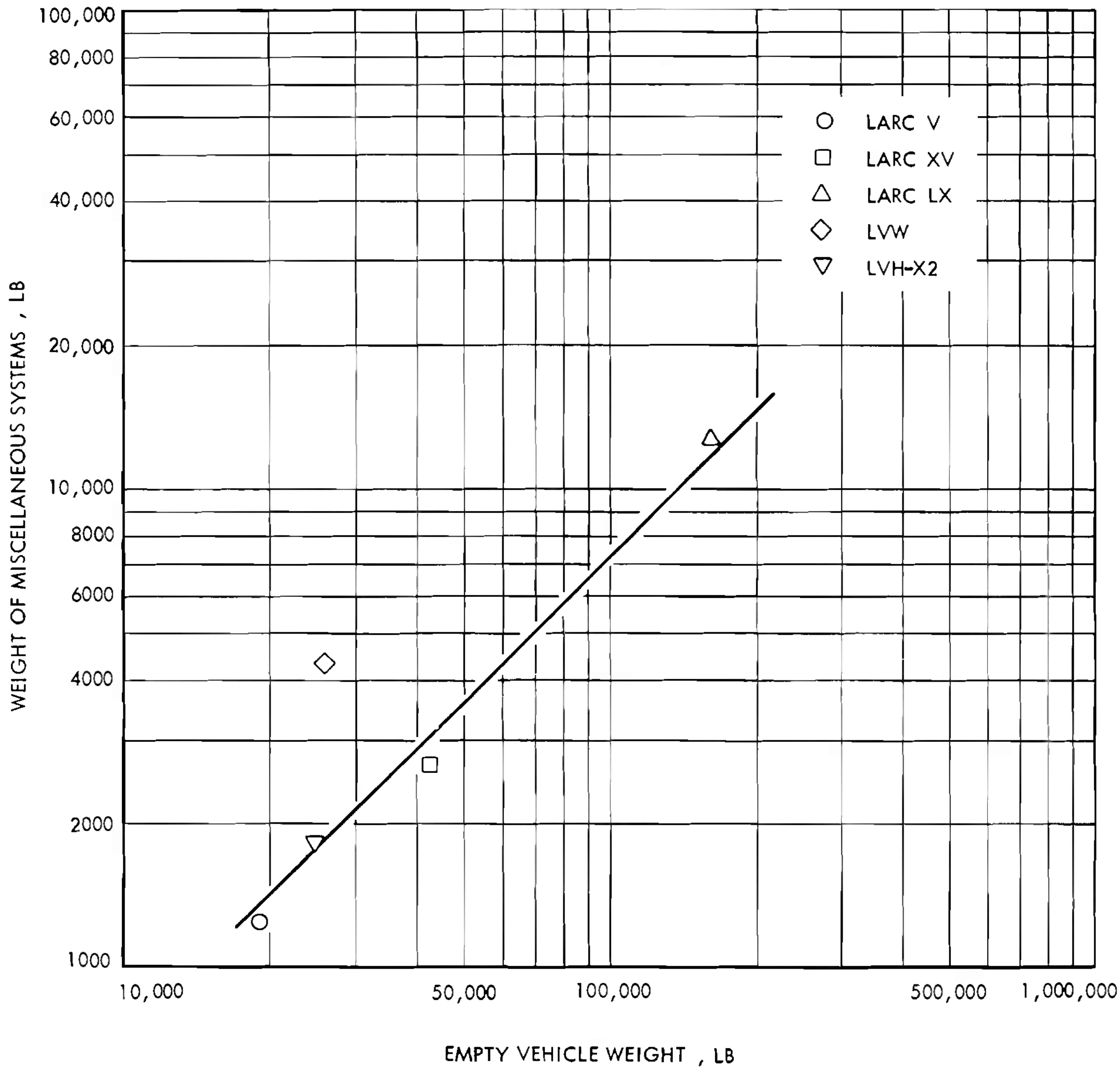


Figure 5-13. Weight Group F—Miscellaneous Systems

5-4.4 MARGINS

Margins are generally added to estimated empty vehicle weight and empty vehicle vertical center of gravity to account for items that are overlooked or underestimated. For a conventional wheeled amphibian on which the conceptual design estimates are based on reliable "as built" data from similar vehicles, these margins can be small. Values on the order of 3 to 5 percent on the empty weight and vertical center of gravity would be reasonable. In the event that components are to be used for which "as built" weight (of identical or similar items) are not available, a larger margin should be added to the appropriate weight groups. In this case the margin should be on the order of 5 to 10 percent of estimated weight of the component. It should be noted that conceptual designs in which weight estimates are based on past designs have some small built-in margin due to advancing technology. This tends to support

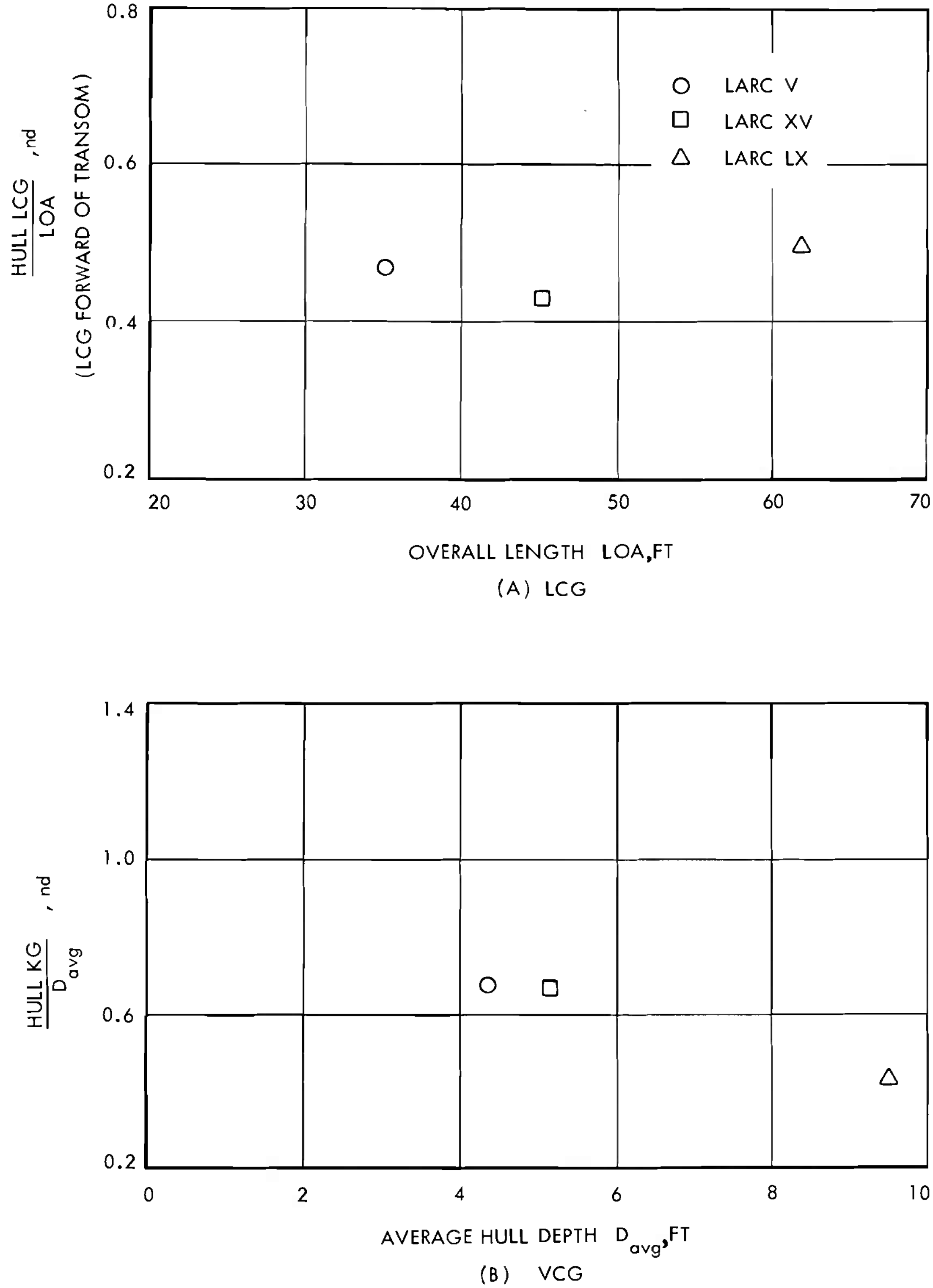
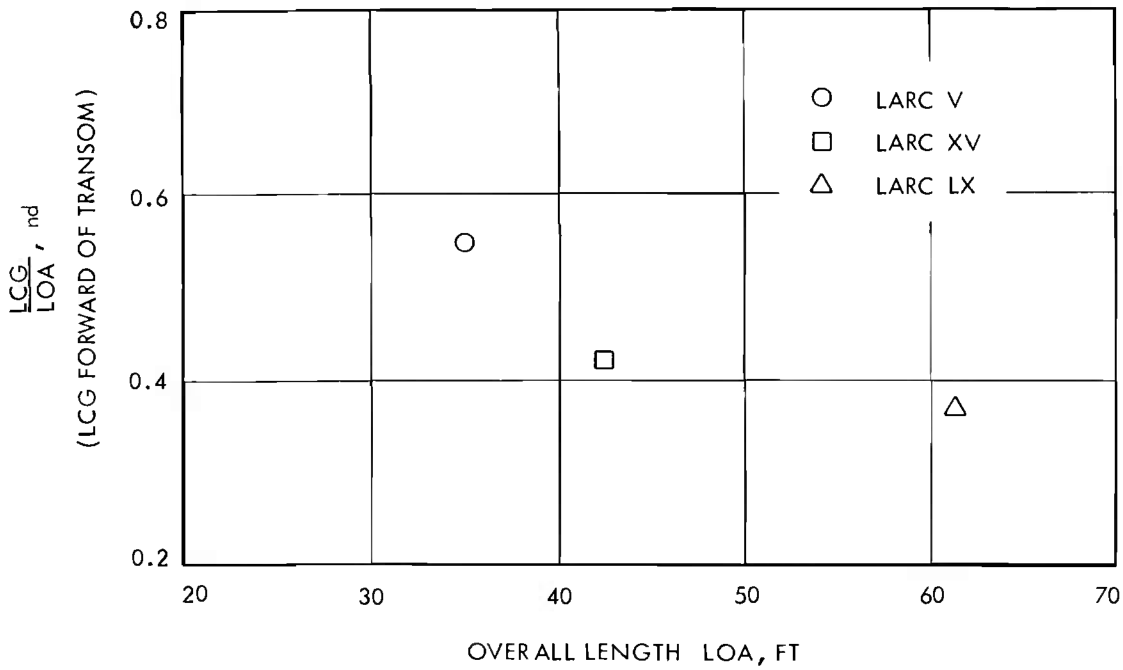
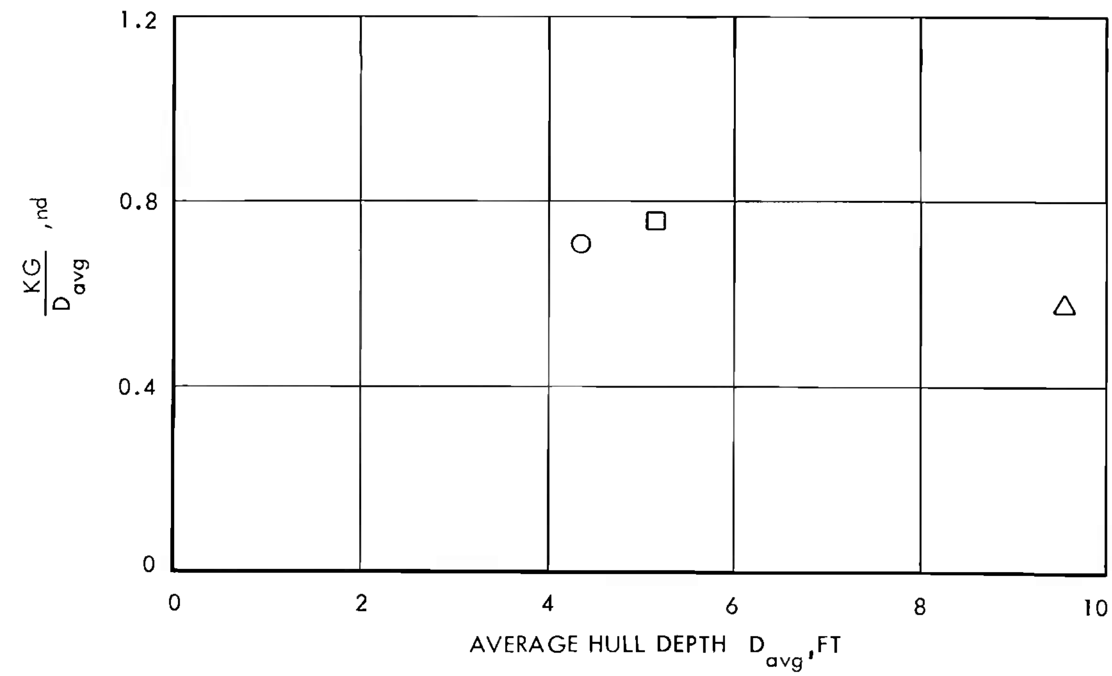


Figure 5-14. Weight Group A—CG Locations



(A) LCG



(B) VCG

Figure 5-15. Weight Group F—CG Locations

the use of small margins during the conceptual designs.

The use of a large or small margin depends to some extent on the design policy and goals. If initial cost is most important, a large margin should be used and if performance is more important, a low margin should be used in association with minimum weight. Design policies or goals of this type should be clearly defined so that in the preliminary and detail design phases, for example, lightweight components are used in spite of higher cost when performance is controlling.

5-5 PERFORMANCE AND POWERING

During the conceptual design phase an estimate of the power required or performance will be made, using the estimated vehicle weight and characteristics. At this stage in the design there is not enough information or time to conduct model tests and detailed propulsor calculations to arrive at an estimate of performance or power required. Accordingly, data from previous designs must be used.

5-5.1 BASIC CONCEPTS

In a true wheeled amphibian the power required to obtain the necessary water speed will almost always be greater than that needed to meet any of the land performance requirements. As a result, during the conceptual design the power required is selected on the basis of waterborne performance unless some very unusual land performance requirement is given. However, the land performance, and especially off-road mobility, affects the geometry of the hull and the size and arrangement of the running gear. This in turn affects the power required to meet the waterborne performance. Thus, consideration during conceptual design must also be given to those aspects of land performance involving mobility.

5-5.2 HULL AND APPENDAGE RESISTANCE

The first step in determining the power required is to estimate the resistance of the amphibian at the design speed and condition. The fundamental principles involved in wheeled amphibian resistance and data on past designs are presented in pars. 7-4.1 through 7-4.1.3. This material should be reviewed and understood before any attempt is made to calculate resistance for a conceptual design.

The procedure available for the calculation of resistance is the selection of the appropriate drag coefficient from Figs. 7-55 and 7-56. The drag coefficient for the faired hull shape can be obtained from Fig. 7-55, and the drag coefficient for the wheels and wheel wells can be obtained from Fig. 7-56. In this way it is possible to account for combinations of hull forms and running gear sizes, and arrangements that have not been tested. The resistance is then given by

$$R_T = \tfrac{1}{2}\rho V^2 \left(C_{D_H} A_H + C_{D_w} A_w\right), \text{ lb} \qquad (5\text{-}13)$$

where

R_T = total resistance, lb
ρ = mass density of water, slug/ft^3
V = forward speed, ft/sec
C_{D_H} = drag coefficient of faired hull, nd
A_H = projected frontal area of faired hull, ft^2
C_{D_w} = drag coefficient of wheels and wheel wells, nd
A_w = exposed frontal area of wheels, ft^2

In the event that the design speed point is not in calm water, the procedures and data presented in par. 7-4.1.2.4 may be used to estimate the added resistance.

5-5.3 PROPULSOR AND POWER REQUIRED

The brake horsepower required by a wheeled amphibian is defined by

$$BHP = \frac{DHP}{\eta_T} + HP_A, \text{ hp} \qquad (5\text{-}14)$$

where

BHP = brake horsepower of the engine, hp
DHP = delivered horsepower to the propulsor, hp
η_T = transmission efficiency between engine and propulsor, nd
HP_A = horsepower required by auxiliary systems, hp

The fundamental principles involved in the calculation of the delivered horsepower to the propulsor are presented in pars. 7-7 through 7-7.6.4. These paragraphs should be reviewed before an estimate of *DHP* is made for a

conceptual design. In the event that the amphibian is propelled with a screw propeller, the procedure which follows may be used to obtain a preliminary estimate of the required delivered horsepower.

In order to make a preliminary estimate of *DHP*, the power required by the propulsor to achieve the desired design speed, the designer must obtain appropriate values of:

a. Drag R_T and effective horsepower *EHP* at the design speed V_D

b. Thrust deduction t and wake fraction w

c. Relative rotative efficiency η_r and the propeller diameter D

Fig. 5-16 is a nomograph which may be used to make a quick estimate of propeller open water efficiency η_o which is needed to find the necessary delivered horsepower *DHP*. This nomograph is based on the assumption that the propeller efficiency is 80 percent of the ideal efficiency η_i predicted by simple momentum theory (see par. 7-7.4.1). The 80-percent assumption represents the state-of-the-art in marine screw propeller design for carefully designed "optimum" propellers. Thus, the efficiency found from the nomograph will almost always be on the high, i.e., optimistic, side. It is suggested that the nomograph value be decreased by 3 or 4 percentage points to obtain a more realistic value of propeller efficiency for amphibians. Use of the nomograph will be illustrated by the following example problem using data for a model LARC V from Table 7-35.

Example

Given:

V_D	= 9 mph	η_r	= 0.955
D	= 2.5 ft	EHP	= 70 hp
$(1-t)$	= 0.647	R_T	= 2901 lb
$(1-w)$	= 0.748		

Eq. 7-82:

$$V_a = V_D\ (1 - w)1.47 = 9(0.748)1.47 = 9.9 \text{ ft/sec}$$

Eq. 7-83:

$$T = R_T/(1 - t) = \frac{2901}{0.647} = 4500 \text{ lb}$$

A line is drawn between the values of V_a and D on the appropriate scales. A second line is drawn between point "A" and the value of T on the thrust scale. The value of η_o = 0.38 is read (compared to 0.33 from model tests). If this value is reduced by 0.04, as previously suggested to arrive at a more realistic value, the required horsepower may be calculated (Eqs. 7-54 and 7-96)

$$DHP = \frac{EHP\,(1 - w)}{\eta_o \eta_r (1 - t)} = \frac{70(0.748)}{0.34(0.955)0.647} = 250 \text{ hp}$$

In the event that a propulsor other than a screw propeller is selected, an estimate of the *DHP* may be developed from the information in pars. 7-7 through 7-7.6.4.

The transmission efficiency depends on the type of transmission, the arrangement of the drive train, and the number of gear reductions if the system is mechanical. For existing wheeled amphibians, the approximate transmission efficiency of the marine drive ranges between 90 and 95 percent. This assumes that any torque converters in the drive train are mechanically locked during waterborne operation.

Auxiliary system requirements include the power which must be supplied by the engine to run electrical, hydraulic, and cooling systems. Under normal waterborne operating conditions, this load varies between 7 and 15 hp for the LARC amphibians, assuming the bilge pumps are not operating. In waterborne operation the cooling loads are low on the LARC amphibian since they are all fitted with sea water coolers which have no fan load. If an air-cooled internal combustion engine is installed, the cooling load may be as high as 10 to 12 percent of the gross engine brake horsepower.

5-5.4 LAND PERFORMANCE

To develop arrangement sketches and waterborne performance estimates during the conceptual design phase, information on the size and arrangement of the running gear and suspension systems must be developed. These items are a function of the land performance and, in particular, the mobility required. A number of measures of mobility now in use are discussed in par. 7-8 with the values which apply to wheeled amphibians. The calculation of these mobility measures is too tedious to carry out

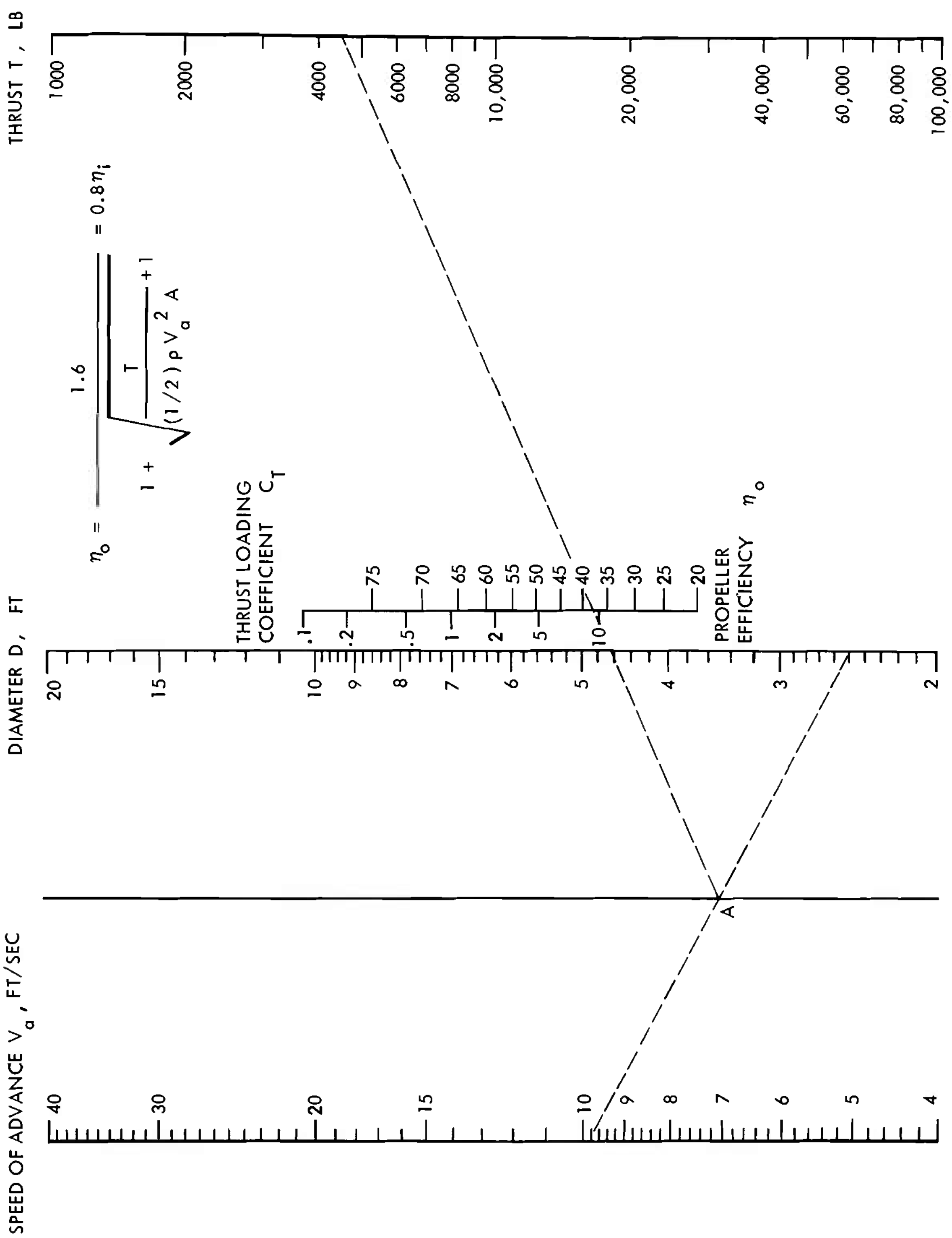


Figure 5-16. Nomograph for Estimating Propeller Efficiency η_o

during the conceptual design iteration process. However, all of these measures are functions which improve as ground pressures are reduced. As a result, it is reasonable to use some measure of ground pressure during the conceptual design to make a preliminary estimate of required tire size. The nominal unit ground pressure and ATAC ground pressure for several wheeled amphibians are presented in Table 5-7.

The ability to cross obstacles is a function of the vehicle geometry and power. Among the most important geometric characteristics for this purpose are the approach, break and departure angles, and the ground clearance. The values of these quantities which have been used on several wheeled amphibians are also given in Table 5-7.

5-6 EXAMPLE CALCULATIONS

In order to illustrate some of the conceptual design procedures and the use of some of the data presented in pars. 5-3 through 5-5.4, an example of a conceptual design calculation is presented. The input conditions were chosen for the purpose of illustration only and do not necessarily reflect current requirements or hardware.

Input:
- Conventional Wheeled Amphibian
- 10-mph full load in calm water
- 10-ton payload *VCG* = 40 in. (Wet Deck)
- Area 265 ft^2, Ramp for unloading
- Endurancc = 8 hr at full power
- Mobility, *NUGP* 10 - 11 psi target for concept design
- Engine - New class of vehicle gas turbine - 1.5 lb/*BHP* 30 *BHP*/ft^2, and 8 *BHP*/ft^3
- Fuel Rate = 0.40 lb/*BHP*-hr

1. Initial Assumptions
   a. Conventional aluminum hull structure

TABLE 5-7 WHEELED AMPHIBIAN LAND MOBILITY

Vehicle	*Approach Angle, deg*	*Break Angle, deg*	*Departure Angle, deg*	*Ground Clearance, in.*		*Nominal Unit* Ground Pressure NUGP, psi*	*ATAC** Ground Pressure p_{av}, psi*
				Hull	*Prop*		
DUKW	35	17	28	17	—	13.1	18.8
LARC V	27	15	20.7	24	16	14.1	20.1
LARC XV	30.5	13.5	22.5	29¼	16½	15.0	21.4
LARC LX	28	14	29	28	—	36.0	51.5
LVW	25	13.5	25	28	—	20.6	29.4
LVH X2	50	15	28	30	—	18.3	26.1

* $NUGP = \dfrac{W}{rb}$

** $p_{av} = \dfrac{W}{0.7\,rb}$

where

W = average load per tire, lb
r = undeflected tire radius, in.
b = tire width, in. (see par. 7-8.4.4)

b. Unsprung suspension, 4 X 4 arrangement of running gear
c. Propulsion—propeller in tunnel
d. Mechanical transmission and drive train

2. Initial Dimensions—
LOA = 40 ft
BOA = 12 ft (overall)

Estimate	Payload/GVW = 0.35
Estimate	GVW = 57,000 lb = Δ
LCB Loaded	18 ft fwd transom

3. Preliminary Tire Size—
Loaded per wheel = 14,250 lb
Tire 27 X 33 in., $NUGP$ = 10.9 psi
Tire Diameter = 87.5 in., width = 30 in.

4. Assume $LWL \approx 36$ ft (at WL),
B = 11.5 ft (at WL),
C_B = 0.70

Mean draft $$H_m = \frac{GVW}{(LWL)B\,C_B\,\rho g}$$

$$= \frac{57{,}000}{36(11.5)0.70(64.4)} = 3.05 \text{ ft}$$

Speed-length ratio,

$$\frac{V_K}{\sqrt{L}} = \frac{10}{\sqrt{36}}\left(\frac{5{,}280}{6{,}080}\right) = 1.45$$

Note: V_K is speed in knots.

$C_{D_H} = 0.47$, $C_{D_w} = 0.40$
(from Figs. 7-55 and 7-56)

5. Hull Area = Beam X Draft
$= 11.5 \times 3.05 = 35 \text{ ft}^2 = A_H$

6. Appendage Area = Radius X Width

$$= \frac{43 \times 30}{144}(4) = 35.8 \text{ ft}^2 = A_w$$

V = 10 mph = 14.7 ft/sec

$$R_T = \tfrac{1}{2}\rho V^2 (C_{D_H} A_H + C_{D_w} A_w)$$

$$= 0.99\,(14.7)^2\,[(0.47\,(35) + 0.40\,(35.8)]$$

$$= 6{,}550 \text{ lb}$$

$$EHP = R_T V/550 = \frac{6{,}550 \times 14.7}{550} = 175 \text{ hp}$$

7. Single Propeller in Kort Nozzle
Propeller Diameter 3.0 ft
$1-t$ = 1.08;
$1-w$ = 1.16;
η_r = 0.93(from Table 7-35)

$$T = \frac{R_T}{1-t} = \frac{6{,}550}{1.08} = 6{,}060 \text{ lb}$$

η_o = 50 percent (from Fig. 5-16)

η_o for amphibian $\approx$ 46 percent

$$DHP = \frac{EHP\,(1-w)}{\eta_o \eta_r\,(1-t)}$$

$$= \frac{175(1.16)}{(0.46)0.90(1.08)} = 440 \text{ hp}$$

8. Assume η_T = 0.95 and Auxiliary Loads = 10 BHP

$$\therefore BHP = \frac{DHP}{\eta_T} + HP_A = \frac{440}{0.95} + 10 = 475 \text{ hp}$$

9. Hull Depth

D ‘midship = Freeboard + Draft
Freeboard = 0.40 X H_m = 0.4(3.05) (Table 5-4)
= 1.2 ft

D ‘midship = 1.2 + 3.05 = 4.25 ft
Freeboard Fore and Aft = 3.0 (midship freeboard)
= 3.0(1.2) (Table 5-4)
= 3.6 ft
Depth Fore and Aft = 6.65 ft

10. Length of Cargo Deck

$$= \frac{\text{Area}}{\text{Beam}} = \frac{265}{12} = 22 \text{ ft}$$

11. A simple sketch of the vehicle profile should now be made for use in making LCG and KG estimates for the weight groups.

12. Weight Estimate
a. Group A—Hull
LOA = 40 ft, BOA = 12 ft , D_{avg} = 5.3 ft

$$\frac{LOA(BOA + D_{avg})}{100} = \frac{40(12 + 5.3)}{100} = 6.9$$

W_A = 12,000 lb (from Fig. 5-9)
KG = 3.40 ft
LCG = 18.8 ft fwd transom (from Fig. 5-14)

b. Group B—Machinery
Specific Weight = 2.25 lb/BHP
W_B = 2.25(475) = 1,070 lb
KG = 3.5 ft
LCG = 31 ft fwd transom

c. Group C—Drive Train
W_C = 2,300 lb (from Fig. 5-11)
KG = 2.0 ft
LCG = 18.0 ft

d. Group D—Marine Propulsor
Specific Weight = 1.75 lb/hp
W_D = 1.75(475) = 830 lb
KG = 1.0 ft
LCG = 5.5 ft

e. Group E—Running Gear and Suspension
To obtain a lower $NUGP$ than on existing amphibians, this design has proportionately larger tires. As a result, the weight of Group E will be proportionately larger. For the LARC V and LARC XV the tires amount to about 6 percent of the gross vehicle weight. A weight estimate for the preliminary tires for this design indicates a weight of about 9 percent of the current estimate of gross vehicle weight. Therefore, to account for the larger tires and rims the weight of Group E indicated in Fig. 5-12 should be increased by 4 percent of the gross vehicle weight.

$\therefore W_E$ = 8,500 + 0.04(57,000) = 10,800 lb
KG = 0.0 ft
LCG = 18.0 ft

f. Group F—Miscellaneous Systems
Fig. 5-13 indicates the weight of Group F will be about 7.5 percent of the empty weight.

$$\text{Empty weight} \approx \frac{W_A + W_B + W_C + W_D + W_E}{1.0 - (0.0075 + 0.03)^*}$$

$$\approx \frac{12{,}000 + 1070 + 2300 + 830 + 10{,}800}{0.895} = \frac{27{,}000}{0.895}$$

$$\approx 30{,}200 \text{ lb}$$

*Note: the denominator accounts for W_F and W_M in the empty weight.

$\therefore W_F$ = 30,200 (0.075) = 2,260 lb
KG = 4.0 ft
LCG = 22.0 ft (Fig. 5-15)

g. Group M—Margins
For this design use a margin of 3 percent of the empty weight.
W_M = 30,200 − (27,000 + 2,260) = 940 lb
KG_{Margin} = 0.25 ft (added to computed KG)

h. Group L—Loads
(1) Payload = 20,000 lb
KG = 7.6 ft (40 in. above cargo deck)
LCG = 18 ft
(2) Crew = 500 lb
KG = 7.0 ft
LCG = 35 ft
(3) Fuel
Rate = 0.4(475) = 190 lb/hr
Endur. = 8 hr
T. Pipe Factor = 0.97 (for fuel in tank below suction inlet)
Weight = 190(8)/0.97 = 1,570 lb
KG = 2.5 ft
LCG = 14 ft

Referring to the Weight and Center Summary following, the calculated weight is about 4000 lb less than the current estimate of 57,000 lb. At this point the iteration should be repeated starting with a new estimate of waterline length, beam, and calculated draft using an estimated gross vehicle weight of between 51,000 and 52,000 lb

For the purposes of this example it is assumed that the iteration has converged with the following characteristics:

LOA = 40 ft
BOA = 12 ft
LWL = 38 ft
B = 11.5 ft
BHP = 450 hp
H_m = 2.56 ft
Δ = 51,000 lb
LCG = 18.4 ft
KG = 3.9 ft

At this point a check of the waterborne stability and trim should be made.

Stability Estimate
$KB = 0.50H_m$
$KB = 0.50(2.56) = 1.28$ ft

WEIGHT AND CENTER SUMMARY

Group	*Weight, lb*	*KG, ft*	*Vertical Mom., ft-lb*	*LCG, ft*	*Horizontal Mom., ft-lb*
A	12,000	3.4	40,800	18.8	225,600
B	1,070	3.5	3,750	31.0	33,170
C	2,300	2.0	4,600	18.0	41,400
D	830	1.0	830	5.5	4,560
E	10,800	0.0	0	18.0	194,400
F	2,260	4.0	9,040	22.0	49,700
Subtotal	29,260	2.02*	59,020	18.7*	548,830
M	940	0.25	—	0.0	—
Empty Vehicle	30,200	2.27	68,500*	18.7	565,000*
Payload	20,000	7.6	152,000	18.0	360.000
Crew	500	7.0	3,500	35.0	17,500
Fuel	1,570	2.5	3,920	14.0	22,000
Gross Vehicle	52,270	4.35*	227,920	18.4*	964,500

* Computed from relation between weights and moments, e.g., 59,020/29,260 = 2.02 (see Eq. 6-1).

$$\nabla = \Delta/(\rho g) = 51{,}000/64.4 = 790 \text{ ft}^3$$

$$I = C_{I_T} B^3 (LWL)$$

$$C_{I_T} = 0.060 \qquad \text{(Table 5-3)}$$

$$I = 0.060(11.5)^3 38.0 = 3{,}460 \text{ ft}^4$$

$$BM = \frac{I}{\nabla} = \frac{3{,}460}{790} = 4.38 \text{ ft}$$

$$GM = KB + BM - KG \qquad \text{(Eq. 5-5)}$$
$$= 1.28 + 4.38 - 3.9$$
$$= 1.76 \text{ ft}$$

Based on the data in par. 7-1.6.2 this is about the minimum acceptable value of *GM*.

Trim Estimate

$$LCB = 0.495LWL \qquad \text{(Table 5-5)}$$
$$LCB = 18.8 \text{ ft}$$

$$GM_{\ell} \approx BM_{\ell} = \frac{11.5(38)^3}{12(790)} \qquad \text{(Eq. 7-28)}$$
$$\approx 66 \text{ ft}$$

$$MT1 = \frac{\Delta GM_{\ell}}{12LWL} \qquad \text{(Eq. 7-29)}$$
$$= \frac{51{,}000(66)}{12(38)}$$
$$= 7{,}380 \text{ ft-lb}$$

$$\text{Trim} = \Delta(LCB - LCG)/MT1 \qquad \text{(Eq. 5-9)}$$
$$= \frac{51{,}000(18.8 - 18.4)}{7{,}380}$$
$$= 2.8 \text{ in. by stern}$$

This is a reasonable trim for a wheeled amphibian.

At this point a three view sketch of the wheeled amphibian should be prepared and used to check the feasibility of the assumed arrangement. A sketch of the type is shown in Fig. 5-17. The iteration loop can now be repeated as required.

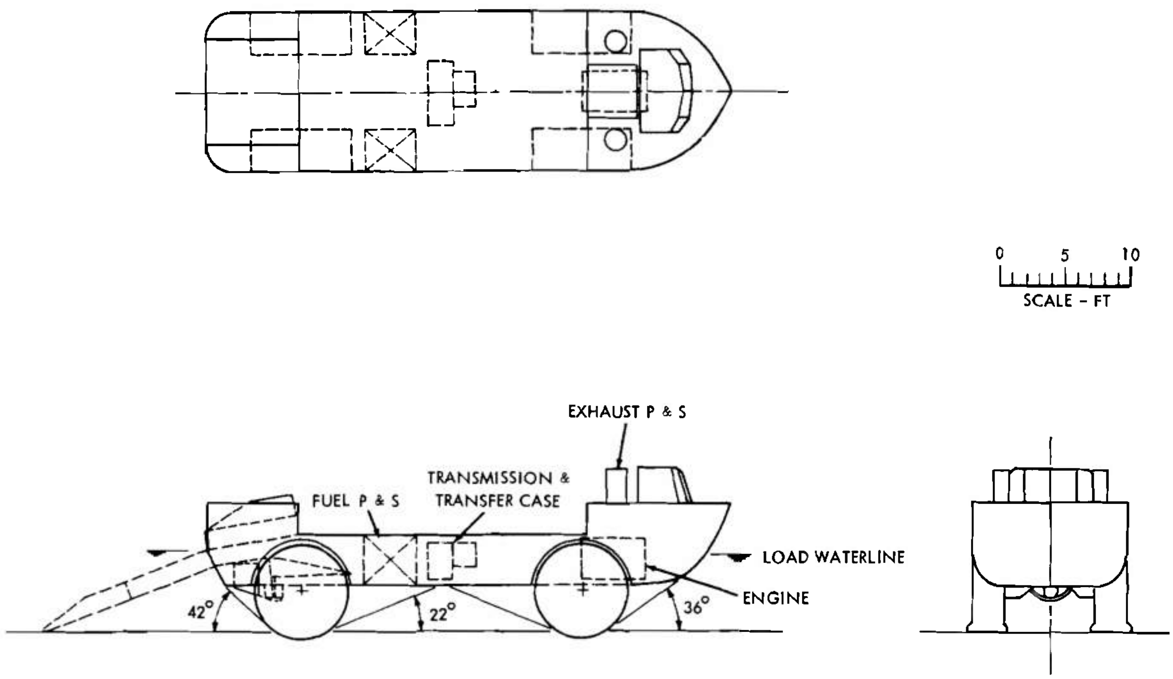


Figure 5-17. Concept Design Sketch

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CHAPTER 6 PRELIMINARY DESIGN

6-1 OBJECTIVE

6-1.1 BASIC PURPOSE

The basic purpose of the preliminary design phase in the development of a wheeled amphibian is to confirm the characteristics developed in the conceptual design and define the major components. This must be carried out to the extent that enough information is made available to complete the final design and working plans.

In addition to leading directly to a final design, the preliminary design may have several other purposes—similar to those presented in par. 5-1.1 for the conceptual design—including trade-off or optimization studies, large scale system analysis or effectiveness studies, "what if" studies, and the development of specifications for procurement. The preliminary design phase is used for these purposes only when the conceptual design phase cannot be expected to yield information in sufficient detail or with a high enough confidence level. This will generally occur when the design specifications or requirements call for a radical increase in performance or a radical change from past practice.

6-1.2 RELATION TO THE DESIGN CYCLE

The preliminary design phase serves as a transition between the conceptual and final design phases. In the preliminary design the major components are designed or selected and arranged. This procedure confirms that the characteristics and performance developed in the conceptual design can be achieved with practical hardware. The preliminary design phase, therefore, must demonstrate the final technical feasibility of the design and identify technical problem areas as well as requirements for the development of special components. If, during the preliminary design phases, the feasibility is shown to be marginal or many technical problems appear; the design cannot continue to the final phase without considerable risk of failure. At the completion of the preliminary design phase, the amphibian should be sufficiently well defined so that the final design and plans can be developed with routine design procedures.

6-2 PROCEDURE

The preliminary design of a wheeled amphibian is an iterative process. Assumptions and estimates made early in the design phase are later checked and, if found in error, revised until the cycle converges. In this respect the preliminary design phase is similar to the conceptual design phase. However, the assumptions and estimates made during the early part of the preliminary design are based on the conceptual design and as a result should be much better than their counterparts made during the conceptual design. This should reduce the number of times the iteration cycle must be repeated.

6-2.1 TYPICAL INPUTS AND OUTPUTS

The inputs to the preliminary design phase consist of the results of the conceptual design phase (i.e., parametric studies, trade-off studies, and estimated vehicle characteristics), the final requirements or specifications (e.g., a QMR or SDR), and any other goals or constraints on the design. It is expected that, by this stage in the design development, interactions between the conceptual design and the preliminary requirements will have eliminated impractical and conflicting final requirements.

The outputs of a formal preliminary design phase include a design summary, preliminary plans, the design calculations, and a design history. The design summary should include a tabulation of the design characteristics and a description of any special features included in the design. Typical plans or drawings that should be developed during preliminary design are listed in Table 6-1. The design calculations and design history present the background data, design methods, and decisions made during the preliminary design. This information will be of great value during the detailed design and in the development of future designs.

TABLE 6-1 TYPICAL PRELIMINARY DESIGN PLANS

Plan	*Comments*
General Arrangement Plans, Profiles, and Sections	These drawings will show the location and arrangements of all major components in the amphibian
Hull Lines	See par. 6-4.6
Structural Arrangements	These drawings should show the arrangement and scantlings typical of basic structure.
Schematic System Layouts	These drawings should show the layout of the control (including steering), hydraulic, electrical, cooling, and any other systems which have primary importance to the amphibian.
Arrangements or Layouts of Specially Designed Components	These drawings should show the preliminary arrangement or layout of any components that must be specially designed.

6-2.2 PRELIMINARY DESIGN ITERATION CYCLE

The iteration cycles for preliminary and conceptual design processes are similar (par. 5-2.2) in that both involve the systematic variation of characteristics until the requirements are satisfied.

The steps in the preliminary design iteration cycle are given in Fig. 6-1 in the form of a flow chart. This flow chart is somewhat of an idealization in that it indicates that the various steps directly follow one another. In general, the press of time and efficient use of manpower will require that several of the steps be conducted concurrently. This implies that effective communications must exist among the members of the design team so that all of the parts "fit" at the end of the cycle.

At the end of the cycle a check will be made against the requirements, goals, and the assumptions on which the preliminary design was based. If these are not satisfied, the iteration cycle must be repeated. Use is made of the data generated in previous cycles to make the process converge. Due to time considerations, it will probably not be possible to conduct more than one set of model tests in spite of the need for several passes through the iteration cycle. However, with the correct choice of test conditions and use of scaling laws, acceptable estimates can still be made (see par. 7-4).

The remaining paragraphs of this chapter cover the topics indicated on the flow chart in Fig. 6-1 as they apply to the preliminary design phase. In some cases the discussions present the general ideas as they apply to preliminary design with the technical details left to referenced paragraphs in Chapter 7. When all of the requirements, goals, and assumptions are satisfied, the design summary and history are prepared.

6-3 PROPULSION SYSTEM DEVELOPMENT

For the purpose of preliminary design, the items considered in the propulsion system include engines, drive train, running gear, and the marine propulsor. For practical purposes, both the technical and economic feasibility of an amphibian design depend on the availability of these components or the ability to design and produce them. As a result, during the preliminary design, suitable candidate components must be selected or a preliminary design for them prepared. The decision to use an existing component or to design a special one depends on considerations of performance, economics, reliability, and maintainability.

6-3.1 CANDIDATE ENGINES

Within the size and speed range that may be expected for future Army wheeled amphibians, there is little chance that the development of a

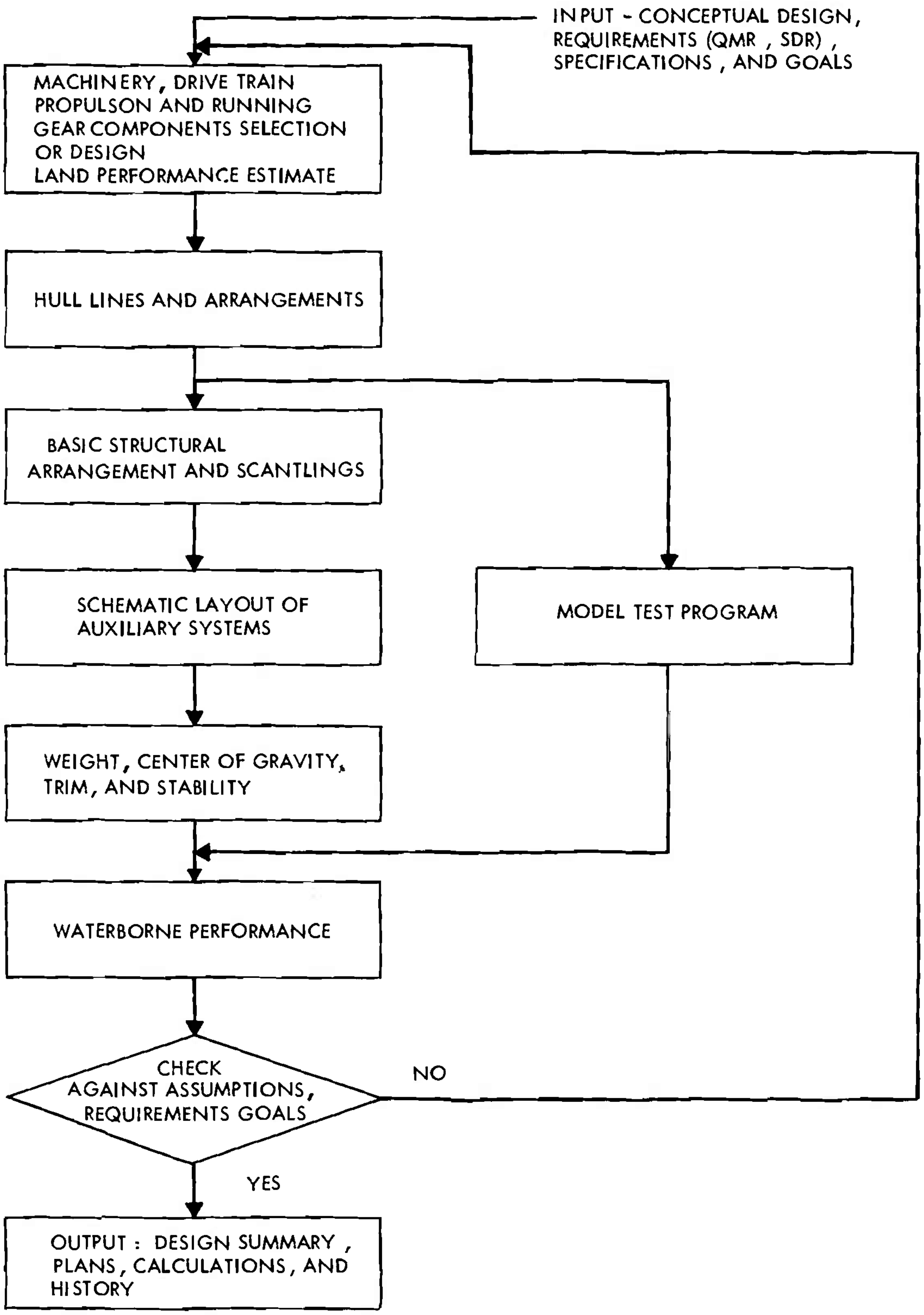


Figure 6-1. Preliminary Design Flow Chart for Wheeled Amphibians

special engine could be justified. Accordingly, some available military or modified commercial engine must be used. This should not be a serious limitation since there is a reasonable variety of possible engines available, unless a water speed in excess of 15 mph is required.

The choice of the actual engine will have a substantial effect on the characteristics, cost, reliability, and maintainability of the amphibian. It is advisable to consider in detail, during the preliminary design, as many engines as possible that come near to meeting the performance requirements. This group of candidate engines should be selected on the basis of power output, size, and weight. The target values for these criteria are obtained from the conceptual design, or previous design cycles through the preliminary design iteration. The group of candidate engines is then narrowed down to a final selection on the basis of the other important criteria such as safety, cost, reliability, and maintenance.

6-3.2 WATER PROPULSOR

Within the context of the preliminary design of the propulsion system, the water propulsor must be defined in terms of type, power and torque required, rpm, and size. The data and procedures needed to generate this information are presented in pars. 6-7.1 and 7-7 through 7-7.6.

In general, the overall water propulsion system will be custom designed to suit the particular amphibian. The components of the system may be off-the-shelf or custom designed. The propeller in a tunnel is the simplest water propulsor in terms of its components. The propeller itself may be a stock item or custom designed, depending on the performance required. The shafts and struts will be arranged to suit the installation.

A waterjet marine propulsor is next in complexity to the simple propeller in a tunnel. A number of complete waterjet units are available as stock commercial items. In the event those available are not suitable, a custom waterjet system could be designed which makes use of an available commercial pump. In this case it would be necessary to design the inlet, exhaust nozzle, and controls. Further, the pump case must be made of lightweight material (aluminum) instead of the cast iron or steel ordinarily used in commercial practice.

A right angle drive marine propulsor will be the most complex type that is likely to be used on a wheeled amphibian. Only a few units are commercially available at this time (1968) that would be suitable for a wheeled amphibian. Even these units would have to be modified to provide a satisfactory retraction system for an amphibian. Although a right angle drive system is mechanically complex, one can be custom designed if justified by the performance requirements. This was the case for the Mobile Floating Assault Bridge-Ferry.

6-3.3 LAND PROPULSION AND MOBILITY

In the preliminary design phase the arrangement of the running gear, the tire size, and torque and rpm requirements at the wheels must be fixed. These performance specifications will define the requirements for these items, and the design cycles through conceptual and preliminary design will provide guidance on arrangements and component selections. Design information and calculation procedures required to make these selections are presented in pars. 6-8, 7-4.2 through 7-4.3, and 7-8 through 7-8.4.

There is a wide variety of military and commercial components available for use in the running gear. Information on these items is contained in the commercial catalogs and other literature published by manufacturers. These components are generally not directly suitable for use on a wheeled amphibian because of the requirement that the running gear must withstand repeated immersion in salt water. Some changes in the materials and seals used in standard components may be required. This was done successfully for the planetary wheel ends used on the LARC series.

6-3.4 POWER TRAIN

The power train transmits power from the engine to the shafting of the marine propulsor to the wheel ends. The gear ratios and torque capabilities of the drive train depend on the engine characteristics, and requirements of the marine propulsor and running gear. These items must be defined before the preliminary design of the power train is undertaken. The same criteria

and general performance requirements apply to the preliminary and final selection or design of power train components. These criteria and requirements are discussed in par. 5-1.3.

In the preliminary design phase the major decision that must be made with regard to the power train is whether to select an available military or modified commercial transmission or to design a special transmission. This decision depends on considerations of performance, size, weight, cost, reliability, and maintenance. These factors, as they apply to this discussion and the power train in general, are covered in par. 7-6.3. In the past, wheeled amphibians such as the LVW have been equipped with custom designed transmissions, whereas the LVH-X2 and the Mobile Floating Assault Bridge-Ferry have been fitted with modified commercial transmissions.

6-3.5 MAINTAINABILITY AND RELIABILITY CONSIDERATIONS

The success of a wheeled amphibian, when placed in the hands of its users, largely depends on its ability to be operated reliably and be easily maintained. This should be uppermost in the mind of the designer when the preliminary selection or design of components in the propulsion system is undertaken. The requirements on the operating life, conditions, and maintenance goals of the amphibian will be given in the QMR or other performance specification. This information must be used in evaluating existing components for possible use or in establishing design factors and goals for custom designed components. It should be noted that maintainability considerations strongly favor the use of components that are already in the U. S. Army supply system. The subject of maintainability and reliability as applied to the design of Army materiel is covered in Ref. 1, and the designer should be familiar with this AMC Pamphlet before working on the preliminary design phase of the propulsion system.

A note of caution is in order here concerning the use of standard components. The virtues ascribed to the use of standard military components in par. 2-9.4 must not lead the designer to the conclusion that their use is a panacea · for all the problems in amphibian design. In the preliminary and conceptual design phases, it is more important to consider new "experimental" items and components than to doggedly cling to old "standard" items. In this way, progress in the evolution of improved amphibians is assured. It is important to remember that so-called "standard items" in our current inventory of amphibians were "experimental items" only a short time ago.

The function of a development cycle is to bring a feasible concept to an optimum or "standard" production stage. In many cases the maintainability and reliability of a standard component may be just as undefined as that of an experimental component because of the unusual amphibian environment. Thus, the final analysis of which is best can only be made after testing has occurred. In any case, the idea of using standard military components simply because they have known properties and predictable reliability is a nonprogressive, nonproductive, and often uneconomic approach.

6-4 INITIAL HULL LINES AND ARRANGEMENTS

For the initial drawings, the lines and arrangements are considered together since the hull form is designed around the machinery, working, and cargo spaces.

6-4.1 ENGINEERING APPROACH

For low water speed amphibians the general approach will be to position all of the components and working spaces with minimum clearances, and then design a minimum volume hull form around the package. This will result in the smallest practical vehicle, which is generally one of the objectives. All designs will have compartments or areas that have extra volume over and above that apparently required. These spaces, frequently at the ends of the vessel, are necessary to meet the buoyancy, stability, resistance, or other hull design requirements. As the design is progressively checked, there will always be conditions which require changes in the dimensions. A certain amount of adjustment will normally be required to arrive at a compact arrangement.

6-4.2 DIMENSION LIMITS AND GOALS

The primary dimension limits and goals should be established during the conceptual

design phase as discussed in par. 5-3. The remainder must be determined during the preliminary design. The limits will be firmly established at the start of the preliminary design process. Many limits determined earlier during the conceptual design stage will have to be checked. The sources of some of the limits are listed in Table 6-2.

6-4.3 COMPONENT SIZES AND WEIGHTS

For each of the potential mechanical or structural components considered, a drawing should be obtained or produced showing the size of the unit and the envelopes of spaces around the unit required for operation, maintenance, and removal. These envelopes should include the space required for tools, to withdraw parts, or to make adjustments.

Accurate weights of the complete units are necessary in order to make satisfactory weight estimates.

6-4.3.1 Sources of Information

The two primary sources of information on component sizes and weights are the component manufacturer and the component user. The manufacturer is trying to sell his products and presents the best possible information and the user has, in general, the history of operation and maintenance for each component.

6-4.3.2 Typical Sizes

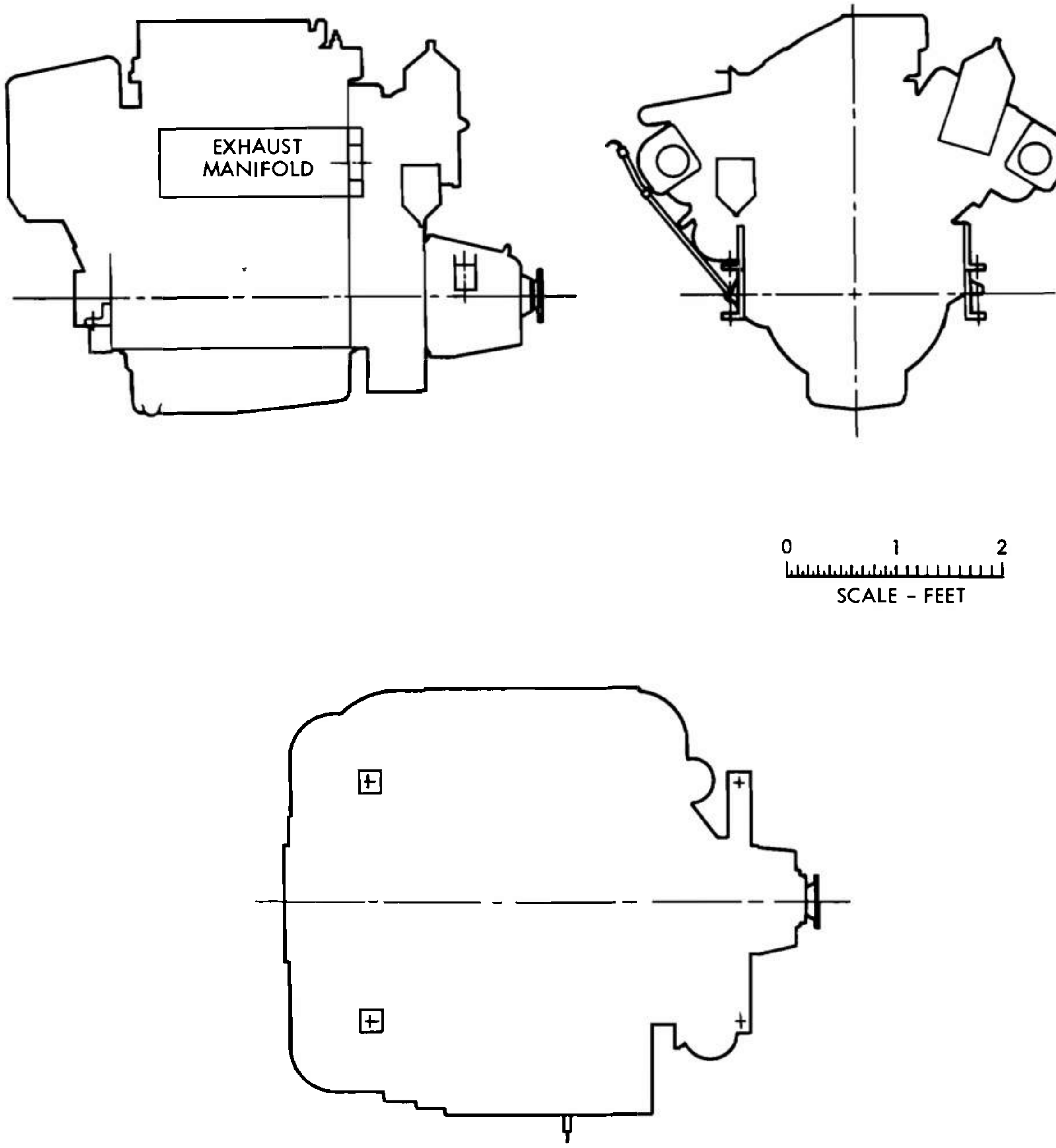
Figs. 6-2, 6-3, and 6-4 are typical drawings of a Diesel engine with the maintenance and removal envelopes drawn. Scale drawings of the components are normally furnished by the manufacturer. The clearance envelopes must be developed by the designer.

6-4.4 BLOCK ARRANGEMENT

The block arrangement is the initial interior arrangement drawing. It uses the envelopes for each component (from par. 6-4.3.2) including the structural scantlings to build up a minimum volume interior arrangement. It is normally matched with the exterior components which have been previously arranged in the conceptual stage. Clearances for construction and maintenance personnel are added where necessary and the minimum size of the hull with respect to the arrangement is complete.

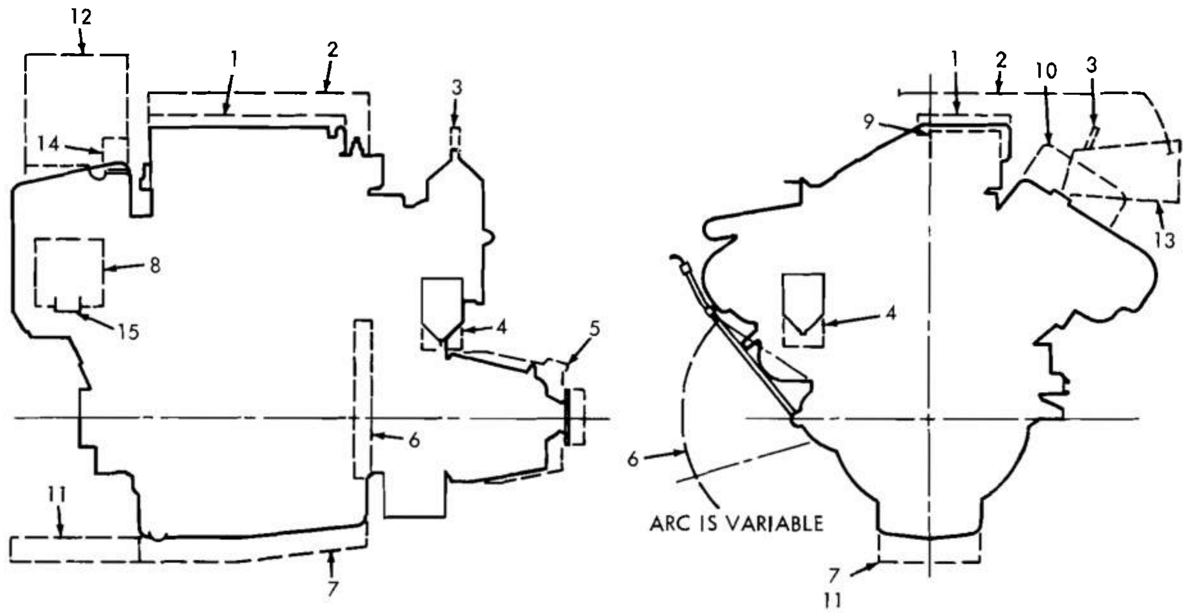
TABLE 6-2 SOURCES OF DIMENSION LIMITS

Dimension	*Limit*	*Source*
Length	Transportability	pars. 2-4.3.6, 2-5; QMR, SDR
Breadth	Transportability and Stability	pars. 2-4.3.6, 2-5, 7-1.6.2; QMR, SDR
Height	Transportability	par. 2-5
Freeboard	Range of Stability and Seaworthiness	par. 7-1.6.2
Wheelbase	Mobility and Maneuverability	par. 2-4.3; QMR, SDR
Ground Clearance and Clearance Angles	Mobility	par. 2-4.3; QMR, SDR
Turning Circle	Mobility and Maneuverability	par. 2-4.3; QMR, SDR; (wheels must be removable)
Machinery Operating Angle	Lubrication	Manufacturers' Data
Driver's Station	Visibility and Protection	par. 7-2



NOTE: SCALE DRAWINGS ARE NORMALLY AVAILABLE FROM THE ENGINE MANUFACTURER. THE ENGINE OUTLINE MAY BE TRACED FROM A SCALE DRAWING AND USED AS A TEMPLATE IN LAYING OUT THE ARRANGEMENT. MANY ACCESSORIES CAN BE MOUNTED EITHER IN OTHER POSITIONS ON THE ENGINE OR ELSEWHERE IN THE MACHINERY SPACE.

Figure 6-2. Typical Marine Engine Outline



GHOST LINES SHOW SPACE REQUIRED TO REMOVE:

1 AIR SILENCER

2 BLOWER (CAN BE REMOVED FROM EITHER SIDE)

3 LUBE OIL FILTER (CENTER STUD TYPE)

4 FUEL FILTER

5 REVERSE GEAR

6 STARTER (ARC OF WRENCH)

7 OIL PAN (DROPPED ONLY, CAN'T BE REMOVED WITHOUT ADDITIONAL CLEARANCE OR REMOVAL OF LUBRICATION OIL PICKUP TUBE)

8 HEAT EXCHANGER CORE

9 GOVERNOR

10 CYLINDER HEAD COVER (ONLY ONE SIDE SHOWN)

GHOST LINES FOR SPACE:

11 FOR LUBRICATING OIL DRAIN CATCH PAN

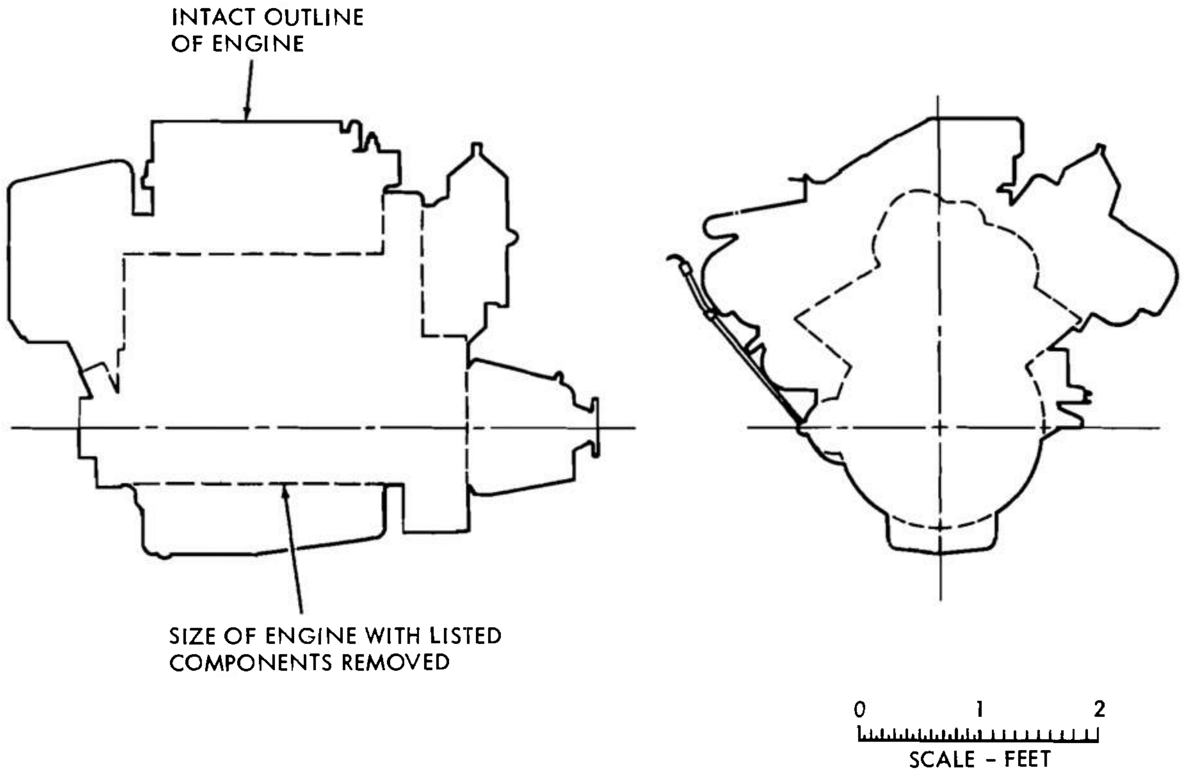
12 FOR HEADROOM FOR WATER LEVEL INSPECTION

13 TO ADD LUBRICATION OIL

14 TO ADD COOLING WATER

15 TO OPEN PETCOCK

Figure 6-3. Typical Envelope of Space Required for Maintenance and In Situ Repairs



LIST OF MAJOR COMPONENTS REMOVED

AIR SILENCER
BLOWER
CYLINDER HEADS AND COVERS
EXHAUST MANIFOLDS
EXPANSION TANK AND PIPING
FUEL FILTER
GENERATOR
LUBRICATING OIL FILTER AND PICKUP TUBE
OIL LEVEL GAGE
OIL PAN
REVERSE GEAR

NOTE: IF ACCESS IS RESTRICTED THE ENGINE MAY BE PARTIALLY DISMANTLED PRIOR TO REMOVAL. AN OUTLINE SIMILAR TO THIS MAY BE USED AS A TEMPLATE BY THE ARRANGEMENT DESIGNER.

Figure 6-4. Typical Stripped Engine Outline

6-4.5 HULL LINES

6-4.5.1 Purpose of Hull Lines

The hull lines drawings serve a number of purposes in the overall scheme of amphibious vehicle design. Fig. 7-1 is an example of an amphibious vehicle lines drawing, and par. 7-1.1.1 includes an explanation of the lines drawing.

The lines drawing graphically describes the shape of the outer hull surface. The hull lines provide the boundaries along the hull length to check space and clearances of various components. From the lines drawing, the builder obtains information on size and shape of plates and frames required to construct the hull. The form calculations relating to vehicle displacement, center of buoyancy, trim, and stability are calculated using the lines drawing.

There are two stages of lines drawings—preliminary and final. The final lines drawing is a product of the final stages of the design cycle, wherein the lines drawing is refined by adding additional stations, waterlines, and buttock lines. The curved surfaces may be made developable during the final design stages if the designer so chooses. The final lines drawing requirements and procedures are given in par. 7-1.1.2.

6-4.5.2 Preliminary Lines Drawing

Preliminary lines drawing is an output of the preliminary design cycle. The design cycle, as explained in par. 6-2, is an iterative process with various vehicle dimensions and characteristics changing with each iteration. In the preliminary stages, the lines drawing may undergo many successive modifications.

Before the preliminary lines drawing can be developed, there are a number of preliminary characteristics which must be known or estimated. These include:

a. Displacement

b. Longitudinal Center of Buoyancy (LCB) location

c. Principal dimensions (length overall, waterline length, beam, draft, hull depth forward, aft, and 'midships, and the approach, beam, and departure angles)

d. Block arrangements

e. Running gear size and characteristics

f. Marine propulsor type and size

g. Hull material and construction procedures

This information is obtained from the requirements, the conceptual design phase, and previous preliminary design work.

6-4.5.3 Lines Development

The following steps may be used in the development of the preliminary hull lines:

a. Select a drawing scale and lay out a grid. For a preliminary design, the drawing need not be larger than 18 to 20 in. long.

b. Draw the vehicle centerline profile on the profile plan. The wheels should also be shown at this time.

c. Draw the projection of the deck at side in the half breadth plan.

d. Draw the stations on the body plan. The shapes of the stations are determined by the requirements of the block arrangements plan, the wheels and marine propulsor, buoyancy, and longitudinal center of buoyancy location. The requirements are checked by comparisons and calculations. Reference should be made to pars. 7-1 through 7-1.6.5 for other criteria and calculation procedures.

e. Draw the waterlines and buttock lines in the half breadth and profile plans.

f. Fair the lines in all three views and make certain that all the intersection points are consistent. This process is described in par. 7-1.1.1.

In the event that the required buoyancy or longitudinal center of buoyancy (LCB) cannot be obtained with reasonable station shapes within the assumed dimensions, the dimensions must be changed. It is best to make small changes in buoyancy by changing the draft. Small changes in the LCB can be made by changing the trim. There are practical limits to such changes and major changes in the buoyancy or LCB will require that all of the principal dimensions be adjusted.

6-5 DEVELOPMENT OF BASIC STRUCTURAL DESIGN

The basic structure, as defined herein, includes the general structural configuration, materials selection, and the major scantlings. It excludes local bracing, doublers, special fittings, small foundations, final joint design, and details.

6-5.1 FORCES

The determination of the forces which the vehicle must sustain are discussed in par. 7-3.1. For this phase it is necessary to determine critical forces for the selection of the shell and framing scantlings for all materials under consideration. Smaller amphibians, in almost all cases, will be designed from local loading considerations.

6-5.2 MATERIALS

There are currently only three practical materials for the construction of amphibians, viz., steel, Fiberglas, and aluminum. It is anticipated that the high performance required of wheeled amphibians will usually result in the selection of aluminum for the hulls of future vehicles. Aluminum is discussed in detail in par. 7-3.2. Since rivets tend to leak after aluminum has been heavily stressed, only all-welded construction is recommended for the hull. Hot forming of high strength aluminum without losing material strength is difficult. Therefore, it is preferable to use developable surfaces unless a sufficient number of units will be made so that the cost of special dies is not prohibitive.

6-5.3 STRUCTURAL CONFIGURATION

The structural configuration of a wheeled amphibian must be compatible with:

a. Vehicle design objective and goals
b. Hull material
c. Vehicle configuration
d. Machinery arrangement
e. Anticipated production methods and quantity

Par. 7-3 discusses these and other items in more detail.

In general, the hull of a wheeled amphibian takes all of the loads applied to the vehicle without the aid of a separate frame such as used in a truck. In order to make the most efficient use of the hull material, the structural configuration should be of the stressed skin type. In this configuration the shell plating or skin takes the overall bending and torsional loads. The stability of the skin under these loads is maintained by the direct framing. The local loads are distributed by the skin to the direct framing which further distributes the loads to the indirect framing. For most conventional type wheeled amphibians it may be expected that the local loads will determine the scantlings of the skin, direct framing, and indirect framing.

6-5.4 DESIGN OF BASIC HULL STRUCTURE

The design objective is normally to define a minimum cost structure to meet a given weight allowance and mission requirements. Pars. 7-3.3 through 7-3.4 present details of structural design procedures. In the preliminary design phase it is possible to use structural data from past successful designs for first estimates of structural arrangements (i.e., frame spacing, proportions, and scantlings). Data of this type are presented in Tables 6-3 through 6-5 and Figs. 6-5 through 6-7. These data may also serve as a useful check on the final structural design to confirm that it is reasonable.

6-6 WEIGHT AND CENTER OF GRAVITY

In the conceptual design phase, weights and centers of gravity are generally estimated using previous design experience tempered with engineering judgment and a minimum of calculations. The required horsepower and running gear are dependent on the weight of the vehicle. These two items are a significant part of the weight of the vehicle so that errors in weight estimates tend to lead to large errors in the overall design. It is necessary, therefore, in the preliminary design phase to carefully estimate the weight by groups.

6-6.1 SYSTEMATIC WEIGHT ESTIMATE

In order to use past designs to the best advantage, it is necessary to use a standardized weight breakdown. If the weight classification is logical and used by the industry, the information developed by different organizations can be used by all designers. Since this type of breakdown is not available, Table 6-6 presents a recommended system. This list was developed from the U. S. Navy's weight classification system for boats (Ref. 4).

Figs. 5-9 to 5-13 present weights of previous amphibians in the recommended weight classification system. A further advantage of a systematic weight classification system is that it serves as a reminder of all of the components

TABLE 6-3 TYPICAL ALUMINUM BOAT SCANTLINGS (Ref. 2)

Note: The Table is arranged for longitudinally framed vessels. "Transverse" and "Longitudinal" can be interchanged for a transversally framed vessel.

	Length Range, ft			
Frame Spacing	*24-30*	*30-40*	*45-55*	*60-70*
Bottom transverse	30	36	48	30-48
Side transverse	30	36	48	30-48
Deck transverse	12	12	12	12
Bottom longitudinal	12	10-12	12-18	9-12
Side longitudinal	12	10-12	12-18	9-12
Bulkhead vertical	12	12	12	12
Transom vertical	12	12	12	12
Transom horizontal	24	24	30	30
Fenders 1/2 pipe***	2	3	4	5
Plating	*24-30*	*30-40*	*45-55*	*60-70*
Bottom forward 0.8	0.140 – 0.160	0.190	1/4	5/16
Bottom aft 0.2	0.140 – 0.160	0.190	5/16	3/8
Side	0.130 – 0.160	0.160	3/16	1/4
Deck	0.125	0.125	3/16	3/16
Transom	0.130 – 0.160	0.160	1/4	5/16
Cabin*	0.125	0.125	0.125	0.125
Keel	1/2 X 3	3/8 X 5	3/8 X 6	1/3 X 7
Stem	1/2 X 3	3/8 X 5	3/8 X 6	1/2 X 7
Chine**	1/2 dia bar	5/8 dia bar	3/4 dia bar	1 dia bar
Bulkhead	0.125	3/16	3/16 - 1/4	3/16 - 1/4
Framing	*24-30*	*30-40*	*45-55*	*60-70*
All sections are flat bars unless noted.				
Transverse side	3 X 1/4	3 X 5/16	4 X 3/8	6 X 2 X 1/4 flg
Transverse bottom	5 X 1/4	5 X 5/16	6 X 3/8	6 X 2 X 1/4 flg
Transverse deck	1-1/2 X 1/4	2 X 1/4	3 X 1/4	3 X 1/4
Transverse floors	1/4 plate	1/4 plate	1/4 X 2 flg	1/4 X 2 flg
Longitudinal side	2 X 1/4	3 X 1/4	4 X 1/4	3 X 1 X 1/4 flg
Longitudinal bottom	2 X 1/4	3 X 1/4	4 X 1/4	3 X 1 X 1/4 flg
Engine fdn vertical	1/4 plate	1/4 plate	1/4 plate	3/8 plate
Engine fdn horizontal	1/4 plate	1/4 plate	3/8 plate	5/8 plate
Bulkhead stiffeners	2 X 3/16	3 X 3/16	3 X 1/4	3 X 1 X 1/4 flg
Transom vertical	2 X 1/4	3 X 1/4	4 X 1/4	3 X 1 X 1/4 flg
Transom horizontal	5 X 1/4	5 X 5/16	6 X 3/8	6 X 3 X 1/4 flg

Note: *All dimensions are in inches.*

* *Interlocking extrusions can be substituted here.*
** *Special extrusions would be recommended if production warranted.*
*** *Extrusions or formed plate may be more desirable.*
flg = flange
fdn = foundation

TABLE 6-4 SCANTLINGS FOR RECENT ARMY AMPHIBIANS

Configuration - Longitudinal bulkheads connecting to the inboard sides of the wheel wells. Plate and "T" stiffeners used throughout.

Item	*LARC V*		*LARC XV*
Bottom			
Plate-thickness, in.	3/16		0.190
Alloy	5086-H32		5086-H32
Stiffener-spacing, in.	30 (max)		30 (max)
-scantlings, in.	3 X 3 X 1/4 Tee		5 X 4 X 3/8 Tee
-alloy	5083-H112		5083-H112
Indirect framing	Longitudinal Bulkheads		Longitudinal Bulkheads
-spacing, in.	67		104
Remarks -	Approx. 30 in. transverse frames		Transverse frame spacing selected to suit arrangement
Deck	Forward and Aft	Cargo	
Plate-thickness, in.	3/16	0.250	0.250
Type		Diamond pattern B	Diamond pattern B
Alloy		6061-T6	6061-T6
Stiffener-spacing, in.	30(max)	30(max)	30 (max)
-scantlings, in.	3 X 3 X 1/4 Tee		3 X 3 X 5/16 Tee 4 X 3 X 5/16 Angle
-alloy	5083-H112		5083-H112
Indirect framing	Longitudinal Bulkheads		Longitudinal Bulkheads
Spacing, in.	67		104
Spacing to braces, in.			84 Extra longitudinals used to support tires of gun carriage and vehicles: They are also 3 X 3 X 5/16 Tee
Sides			
Plate-thickness, in.	3/16		0.190
Alloy	5086-H32		5086-H32
Stiffeners-spacing, in.	30 (max)		30 (max)
-scantlings, in.	3 X 3 X 1/4 Tee		3 X 3 X5/16 Tee
-alloy	5083-H112		5083-H112
Indirect framing	Deck and bottom		Deck and bottom
Spacing, in.	36		45
Longitudinal Bulkheads (nonwatertight)	33-1/2 in off centerline		52 in.off centerline
Plate-thickness, in.	3/16		0.250
Alloy	5086-H32		5086-H32
Stiffener-spacing, in.	30 (max)		30 (max)
-scantlings, in.	3 X 3 X 1/4 Tee		3 X 3 X 5/16 Tee
-alloy	5083-H112		5083-H112
Indirect framing	Deck and bottom		Deck and bottom
Spacing, in.	36 (max)		45 (max)
Transverse bulkheads and floors (nonwatertight)			
Plate - thickness, in.	3/16		1/4 3/8 in way of wheels
Alloy	5086-H32		5086-H32
Stiffener-spacing, in.	12-3/4		No stiffeners
-scantling, in.	1-3/4 X 2 X 3/16 channel welded toes to plate		
Alloy	5083-H112		
Indirect framing	Deck and bottom		Deck and bottom
Spacing, in.	36		45
Stem, in.	3/4 X 5-1/2 flat bar		
Welding	Continuous and intermittent		Continuous and intermittent
Torsional rigidity	Partial bulkheads and floors		Partial bulkheads, floors and braces between longitudinal bulkhead frames and transverse deck frames

TABLE 6-5 STEEL BEAM PROPORTIONS (Ref. 3)

These beam proportions are recommended for steel merchant ships.

Component	*Minimum depth of beam, in./ft of beam length*	*Minimum thickness of web, in.*
Panting frames (reduce shell plating deflections from heavy seas)	0.6	
Deck girders	0.7	0.01/in. of beam depth + 0.16
Deck girder in water or fuel tanks (to allow for dynamic effects)	1.0	0.01/in. of beam depth + 0.12
Watertight bulkhead girders or web frames	1.0 plus 1/4 depth of slots for stiffeners	0.01/in. of beam depth + 0.12
Web frames	1.5	0.01/in. of beam depth + 0.14
Hold stringers	1.5	
Deep tanks, girders, and webs (to allow for dynamic effects)	1.75	

whose weight must be included so that none are inadvertently omitted.

Experience has shown that weight estimates based on the sum of estimated component weights will tend to be low. Ref. 5 reports on the weight growths of some recent off-road vehicles. Fig. 6-8 is from that reference with data on the LARC V and LARC XV added. It shows conclusively that weight growth will occur and, as a result, a margin should be added in the preliminary design weight estimate to account for this. There are two basic types of margins of interest—the construction margin and the growth margin. The construction margin is intended to account for the difference between the estimated weight and the actual weight of the amphibian. The growth margin is intended to account for increases in weight due to new equipment added to the vehicle later in its life cycle. Of these two, the construction margin is of more importance to an amphibian design.

Both types of margins take the form of a percentage of the estimated empty weight of the amphibian. The size of the margin must be determined by the judgment of the designer and his confidence in the weight estimating data. The trends indicated by the data in Fig. 6-8 indicate that a preliminary design construction margin on the order of 5 to 10 percent of the estimated empty weight would be suitable for a wheeled amphibian.

6-6.2 CENTER OF GRAVITY ESTIMATE

The center of gravity of n components is easily calculated using the following formula

$$X = \frac{\sum_{i=1}^{n} x_i w_i}{\sum_{i=1}^{n} w_i} \tag{6-1}$$

where

X = distance from base plane to the center of gravity, ft

x_i = distance from base plane to the center of gravity of the i^{th} component, ft

w_i = weight of the i^{th} component, lb

This formula is used in the longitudinal, transverse, and vertical directions to completely determine the position of the center of gravity. Recommended base planes are at the stern for the longitudinal direction, at the hull bottom for vertical direction, and at the hull center plane for the transverse direction with forward, up, and starboard being the positive directions, respectively. The transverse center is needed only if part of the arrangement is unsymmetrical. In the arrangement, some allowance is desirable for shifting heavy components fore and aft in the final design in order to adjust the trim. The longitudinal center is used for trim estimates, and the vertical center is used for the stability calculations. Typical longitudinal center of gravity (LCG) and vertical center of gravity (VCG) values from previous designs are listed in Table 5-5 and Figs. 5-14 and 5-15.

Margins for the center of gravity are normally applied only in the vertical direction. For a design, it is usually a fixed distance which is added to the VCG. If an error in trim in one direction would be serious to the design, a margin on the LCG should be added. Normally, errors in either the fore or aft direction are equally bad and a margin is not used. For boats, the U. S. Coast Guard sometimes uses a 6-in. forward margin so that if the estimates are accurate, the boats trim down by the stern.

6-7 WATERBORNE PERFORMANCE

6-7.1 SPEED AND POWER

It is possible that no specific type of propulsor would be selected in the conceptual design stages. In this case, the preliminary design procedures should be expanded to include candidate propulsor types so that a rational selection may be made among them in this stage of the design. The discussion which follows applies to a conventional screw propulsor. However, the design process is essentially similar for any other suitable propulsor, e.g., waterjet.

The principal variables concerning the propulsor design must be selected or at least narrowed down within reasonable limits in the preliminary design stage. Propeller diameter, design rotative speed, blade-area ratio, number of blades, and pitch can be selected from design charts (par. 7-7.5.2) using a preliminary estimate of design speed, resistance, wake fraction, and thrust deduction. In the process, the propeller efficiency and, hence, required horsepower will be calculated. The rotative speed and required horsepower are used in the selection of a suitable power plant and any necessary marine drive reduction units.

The problem may, of course, be worked inversely, i.e., the candidate power plant(s) characteristics may be used to select the optimum propeller and, thus, arrive at a realistic design speed.

The effect of any alternative propulsion systems or design features (e.g., controllable pitch) on the speed and power and on the overall design concept should also be evaluated. If a Kort nozzle propulsor is to be used, the geometry of the shroud must be delineated. The off-design and/or static thrust of each candidate propulsor should be estimated if such performance is deemed important in the selection of a suitable design.

6-7.2 STATICAL STABILITY AND TRIM

The stability and trim of the amphibian are of primary importance and must be checked at every stage of the design. The weights and arrangements are constantly changing in the evolutionary design process. Generally, the gross vehicle weight and vertical center of gravity will tend to rise throughout the design stages. Consequently, subsequent increases in the length and/or beam may be required to insure adequate displacement and stability. Similarly, changes in the arrangements may be necessitated by undesirable trends in the static trim as the design evolves.

Other changes in the design—such as tire size and alterations in the propeller tunnel—will, of course, also affect the displacement, stability, and trim of the amphibian and must be taken into account.

6-7.3 SEAKEEPING AND MANEUVERABILITY

The surf zone always presents the greatest test to an amphibian with respect to seakeeping ability. Thus, the main requirement concerning seakeeping that an amphibian design must

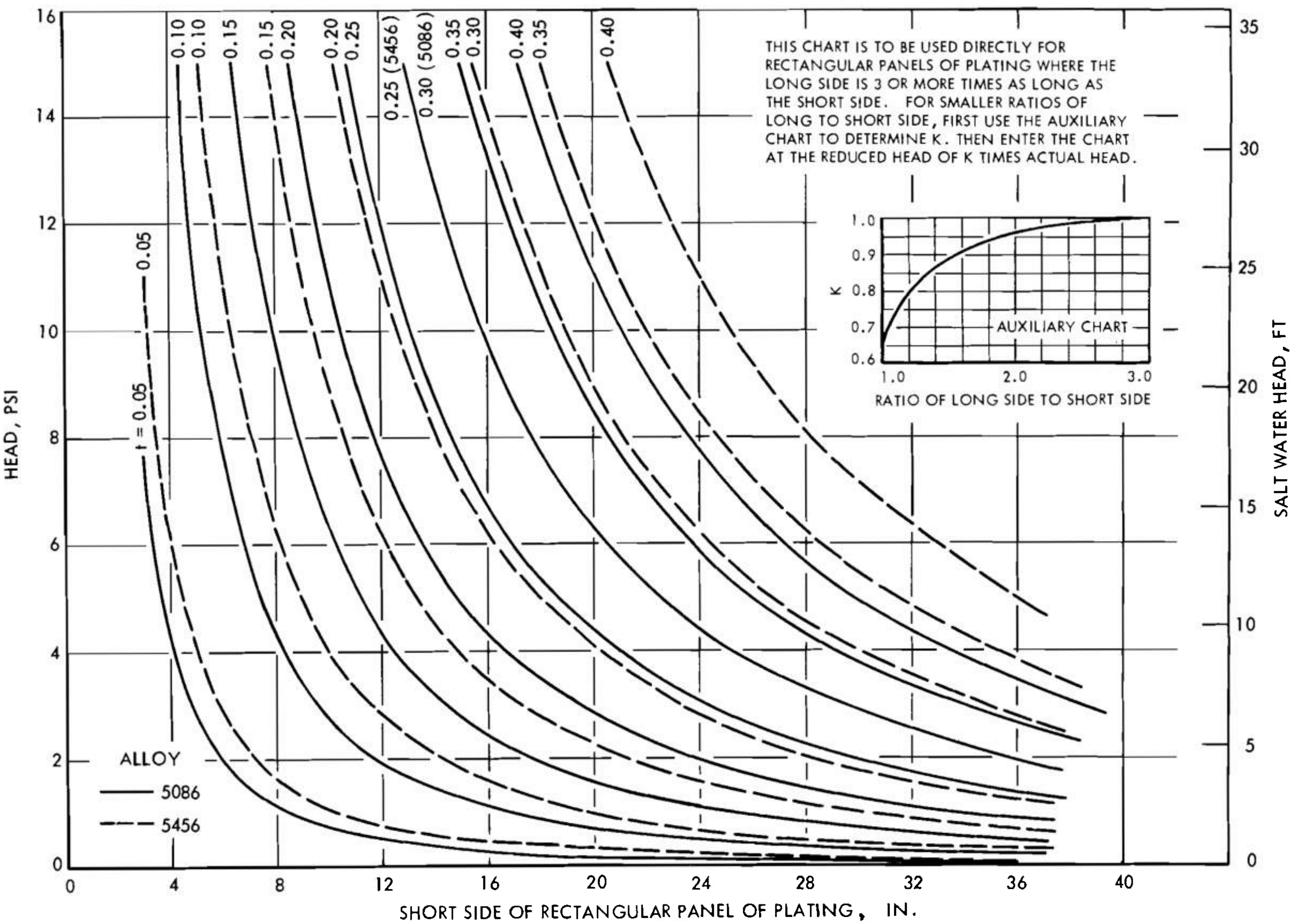


Figure 6-5. *Minimum Thickness of Aluminum Shell Plating With No Permanent Set*

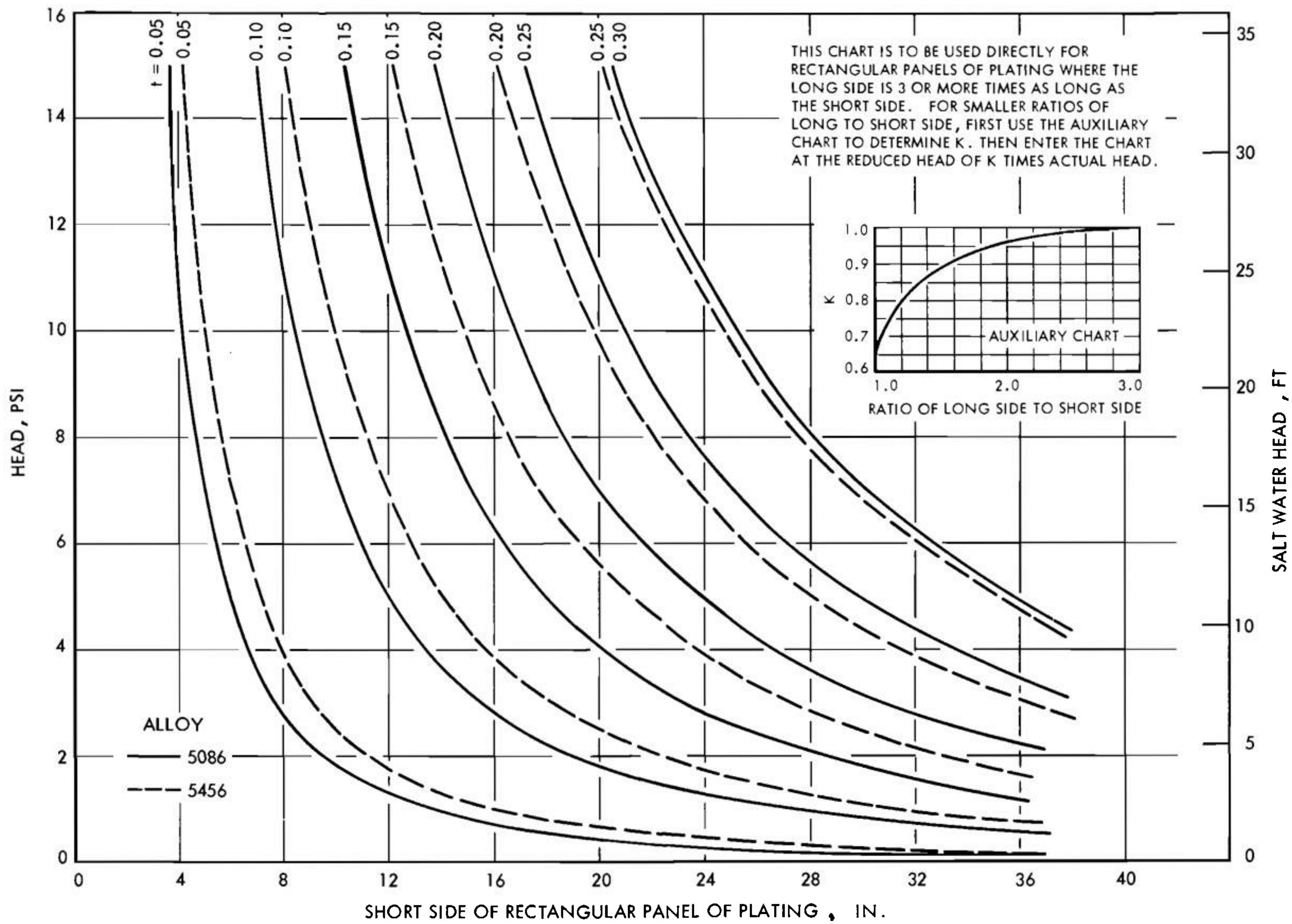


Figure 6-6. Minimum Thickness of Aluminum Shell Plating With Slight Permanent Set < t/2

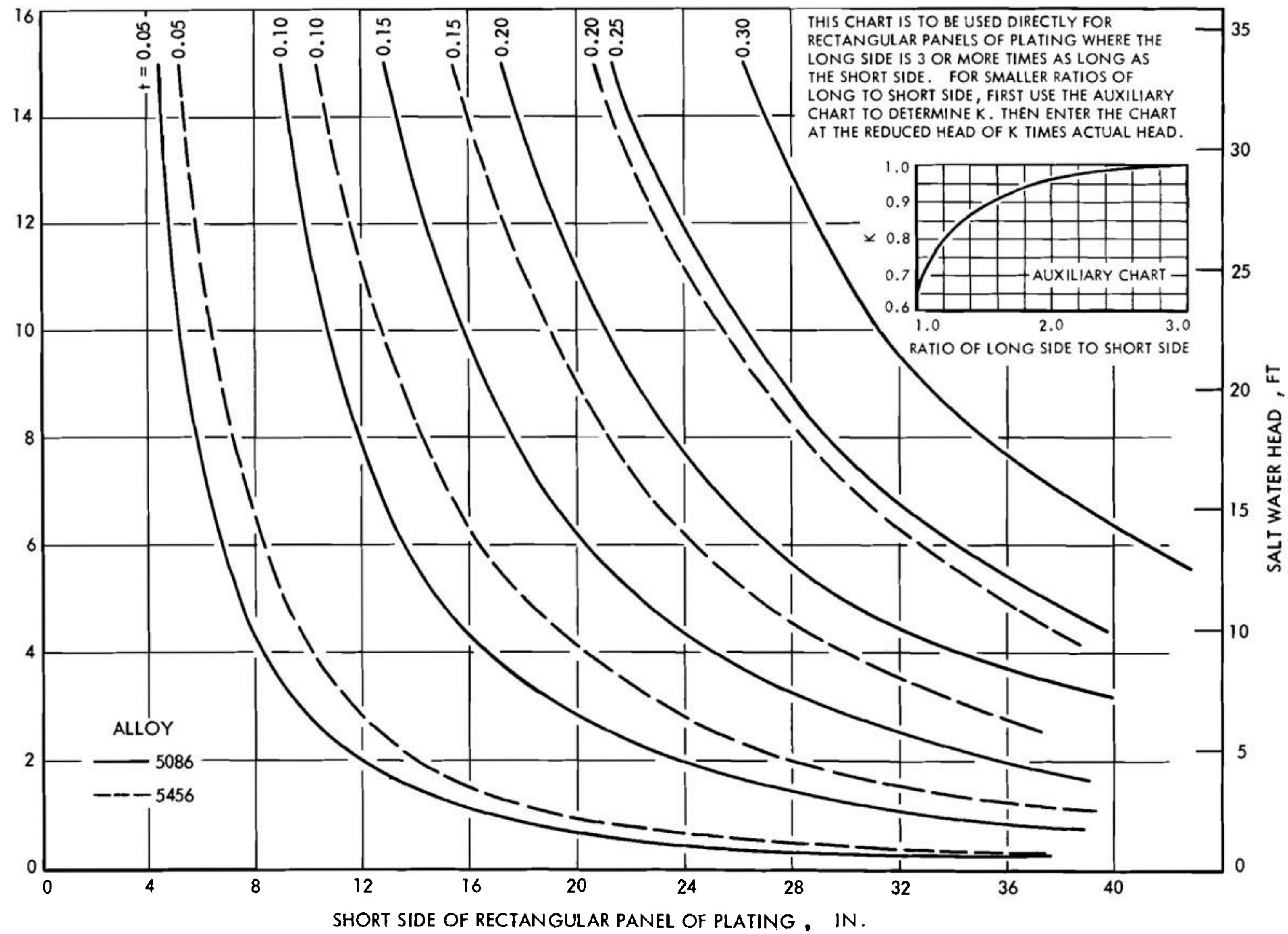


Figure 6-7. Minimum Thickness of Aluminum Shell Plating With Permanent Set > t/2

TABLE 6-6 RECOMMENDED WEIGHT CLASSIFICATION FOR WHEELED LOGISTIC AMPHIBIANS

Major Groups	*Description of Included Items*
A	Hull Structure
B	Machinery
C	Main Drive Train
D	Marine Drive Train
E	Running Gear and Suspension System
F	Miscellaneous Systems
-	Margin
-	Loads

The total of the major groups is equal to the gross vehicle weight (GVW). The total without the cargo and crew is the curb weight.

Hull Structures

A-01	Shell plating—includes:	
	Sides	
	Bottom	
	Transom	
	Headlogs	
	Rub rail (if integral)	
A-02	Framing—includes:	
	Keel and keelsons	Chines
	Gunwale	Stringers
	Stiffeners	Breast hooks
	Web frames	Girders
	Gussets	Brackets
A-03	Flats—includes:	
	Framing	
	Plating	
A-04	Main Deck—includes:	
	Deck beams	
	Toe rail	
	Gussets	
	Plating	
A-05	Cab—includes:	
	Framing	
	Beams	
	Plating	
	Brackets	
	Hatch framing	
	Door framing	
	Windshields	
A-06	Foundations for Main Propelling Machinery—includes:	
	Foundations for shaft logs	
	Propeller struts	
	Foundations for main engines	
	Foundations for inboard/outboard drives	
	Foundations for machinery auxiliaries	
A-07	Foundations for Other Equipment—includes:	
	Foundations for winches	
	Doublers for cleats, chocks, and other deck fittings	

TABLE 6-6 (Continued)

Hull Structures	*(Cont'd)*
A-08	Bulkheads—includes: Stiffeners Plating
A-09	Ballast and Buoyancy—includes: Solid ballast Locked-in-place liquid ballast Permanent flotation material (not including buoyancy tanks)
A-10	Doors, Hatches, Manholes, and Scuttles—includes: Doors, hatches, manholes, and scuttles Ramps Companionways Operating gears for above
A-11	Masts and Spars—includes: Booms Masts Staffs Sockets and foundations
A-12	Fastenings—includes: Mechanical fasteners Welding Adhesives
Machinery	
B-01	Machinery—includes: Main engines (complete with all attached accessories except transmission)
B-02	Main Engine Exhaust System—includes: Piping Smokepipe (outer and inner casing) Lagging (insulation) Mufflers Insulation (sound) Fluids in system
B-03	Main Engine Coolant System (When not Attached to Main Engines) —includes: Piping Hoses Clamps Heat exchangers Keel coolers Coolant wells and gratings Expansion tanks Fluids in system Fans Fan drives Radiators
Main Drive Train	
C-01	Main Drive Train—includes: Transmissions Differentials

TABLE 6-6 (Continued)

Main Drive Train	*(Cont'd)*
C-01	Torque converters Shafting Couplings Fasteners Fluids in system Universals Clutches Hydraulic retarders Transfer cases Keys
Marine Drive Train	
D-01	Marine Drive Train—includes: Shafting Couplings Fasteners Bearings Stuffing boxes Shaft logs Fairwaters Propellers or other propulsors Kort nozzles Struts On-board-water (waterjets)
Running Gear and Suspension System	
E-01	Running Gear and Suspension System—includes: Tires Tubes Wheels Wheel ends Wheel covers Springs Shock absorbers Fasteners Hydraulic steering cylinders Mechanical steering linkages Service brakes Fluids in system Bearings (for axles if separate) Axle boots Clamps Torsion bars Keys Trailing and leading arms King pin assemblies Seals (in system) Parking brakes Wheel retraction system (complete) Control arms Ball knuckles

TABLE 6-6 (Continued)

In general, Group E includes all components of the land gear and suspension system that are connected outboard of the universal joint (if any) at the end of the fixed shafting or differential. Items such as hydraulic steering pumps, or variable tire inflation systems, are, in general, secured to the hull structure and are included in Group F—Miscellaneous.

Miscellaneous

F-01	Control System—includes: Control panels Remote controls
F-02	Fuel Oil System—includes: Piping Valves Pumps (if not part of engine) Strainers Manifolds Tanks (except built-in tanks) Filters Filling connections
F-03	Spare Parts—includes: Spare propellers Spare shafts, keys, nuts Spare bearings Spare engine parts On board tools Spare parts for electronics
F-04	Auxiliaries—Equipment powered by, but not attached to, main propulsion engine and not properly grouped with another distinct group—includes: Hydraulic pump Generators Vacuum pumps Air compressors Central tire inflation systems
F-05	Electric Power Generation and Cable—includes: Generators (independently driven) (including normal operating liquids) Piping Exhaust Foundation fastenings Batteries Battery cable and clamps Battery box Voltage regulators Resistors Panels Cable Wireways Supports and drip pans Switchboards Muffler

TABLE 6-6 (Continued)

Miscellaneous	*(Cont'd)*
F-06	Lighting System—Distribution and Fixtures—includes: Switch gear Cable Supports Wireways Fastenings Lighting fixtures
F-07	Navigational Systems and Equipment—includes: Compasses Repeaters Binnacles Running, signal, and anchor lights Searchlights Blinkers Flags Megaphones Horns
F-08	Interior Communication System—includes: Telephones
F-09	Countermeasure and Protective Systems (except electronics)—includes degaussing system and equipment
F-10	Electronic Systems—includes: Radios Radar Cable Wireways Supports Motor-generator sets Antennas Switch gear
F-11	Ventilation and Heating—includes: Ducting Heaters Cowls and louvers Blowers and fans Brackets
F-12	Piping Systems—includes: Fresh water system Plumbing fixtures Tanks (when not built-in) Hangers Chemical fire system (not portable) Bilge drainage system Pumps Pipe insulating materials
F-13	Steering Gear System—includes: Rudders Stuffing tubes Rudder carrier Steering gear

TABLE 6-6 (Continued)

F-13	(cont'd) Distant controls Sheaves Wire rope or rod Steering wheel Sprocket and chains Brackets
F-14	Winches, Capstans, and Anchor; and Cargo Handling Systems—includes: Winches Windlasses Capstans Anchors Chain or rope, and Grapnel and lines All cargo handling
Does not include fixed items for use with these systems such as bitts, chocks, davits.	
F-15	Hull Fittings—includes warping and towing fittings, and hand rails: Tow posts Tow bridles Davits Chocks (closed or open) Cleats Bitts Fairleads Blocking and brackets Grab rails Hand rails Life rails (wire rope or pipe) Guard rails (wire rope or pipe) Stanchions and sockets
F-16	Boats, Boat Handling, and Stowage—includes: Boats complete (dinghies, etc.) Boat chocks Boat lashings Boats, rubber life (complete) Stowage for life rafts (wood or metal) Boat falls (lines and blocks) Lifting pads (for hoisting) Boat davits
Lifting slings for the vehicle are not included as part of the light condition unless they are normally carried on board the vehicle. These are treated as a load when determining the hoisting condition.	
F-17	Rigging and Canvas—includes: Standing rigging Running rigging (for winches) Wire rope and fittings (for mast) Manila rope Mooring lines Heaving lines Lashings

TABLE 6-6 (Continued)

F-17	(cont'd) Blocks Deck tackle Awnings Weather cloths Canvas covers Cushions
F-18	Ladders and Gratings—includes: Ladder rungs Steps Portable ladders All gratings
F-19	Joiner Work—includes: Bulkheads, minor and divisional Partitions Wire mesh Doors in the above items Tanks and supports when separate from hull structure and not part part of the system
F-20	Painting—includes: Paint Preservatives Cement Other coatings
F-21	Insulation—includes: Insulating materials Sound proofing Sheathing

Does not include pipe lagging which is included with the pipe in piping systems.

F-22	Stowages and Lockers—Lockers and Fittings—includes: Hardware Shelving Bins Stanchions (portable) Gratings Battens Work benches (in machinery spaces) Stowage racks (in machinery spaces) Stowage bins (in machinery spaces) Life jackets Lockers not properly grouped elsewhere
F-23	Furniture—includes all furniture, built-in or otherwise, such as: Chart table Desks Seats Chairs and stools Litter supports Plotting boards

TABLE 6-6 (Concluded)

F-24	Equipment—includes: Signal searchlights (portable) Hand lanterns Medical chests First aid boxes Fire extinguishers (portable) Rope or other fenders (portable) Litters Ring buoys Cleaning gear Fire and bilge pump (portable) Hoses Damage control gear Books Boat hooks Deck gear Buckets Life jackets Mess gear Stoves Hot plates Other on equipment materiel
F-25	Armament—includes: Guns Small arms and fire control equipment Ready service boxes Small arms locker Pyrotechnics locker
Margins	Margins—includes: Growth Construction Conceptual design Preliminary design Detailed design
Loads	Ammunition—includes: Small arms cartridges Pyrotechnics Complement—includes: Men and effects (Passengers are included with cargo) Stores—includes: Dry provisions General stores Medical stores Machinery stores and lubricating oil Potable Water—includes: Drinking water Cargo—includes: All cargo (dry, bulk, or liquids) Passengers and effects Litter patients Fuel—includes: Diesel oil Gasoline

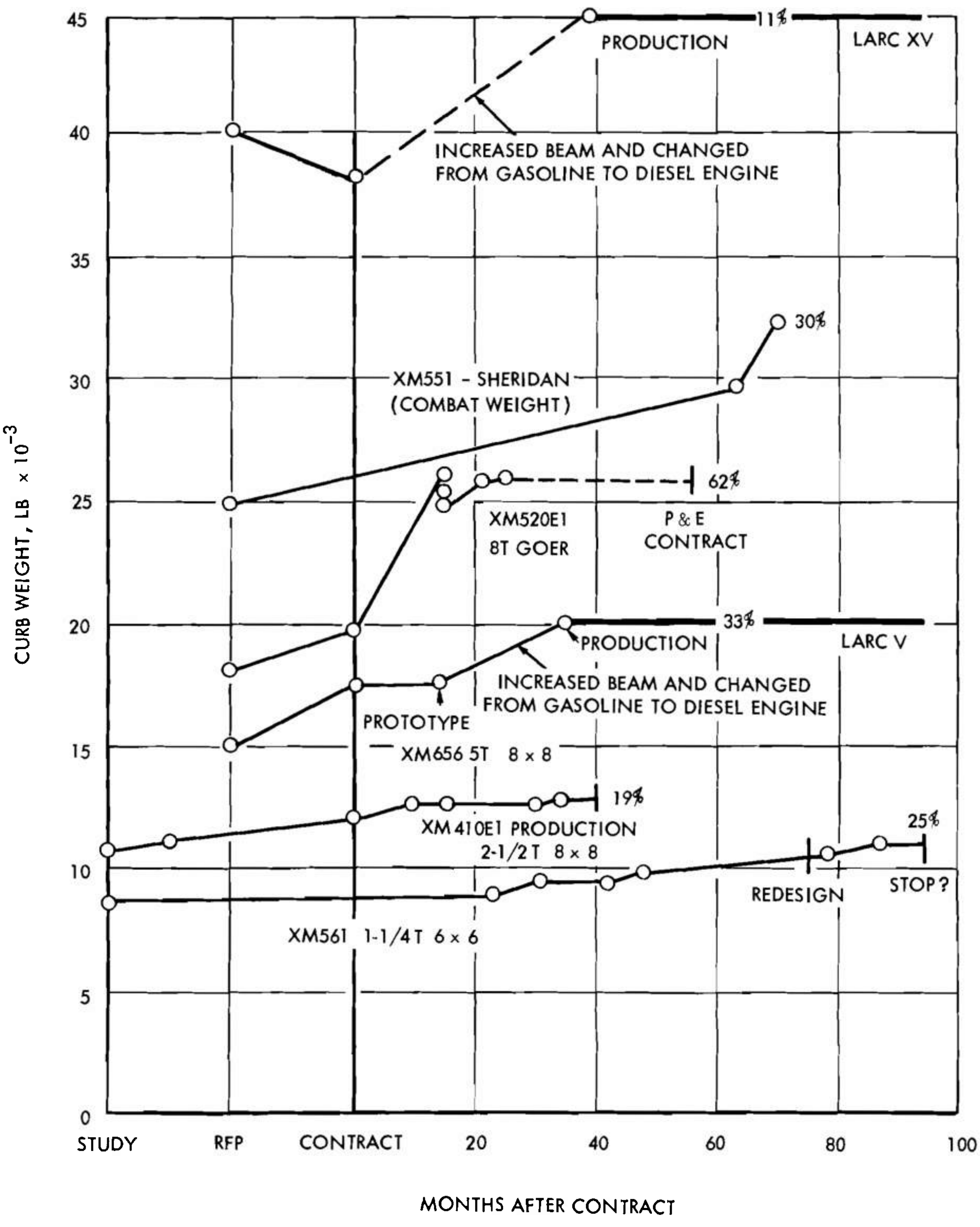


Figure 6-8. Weight Growth of Recent Experimental Off-road Vehicles and the LARC V and LARC XV (Ref. 5)

satisfy is the ability to safely pass through surf whose severity is usually specified in terms of breaker height from crest to trough. Unfortunately, there are no formulas available with which the designer can predict, quantitatively, surfing ability. Model testing does not offer much help in this respect, although qualitative comparisons of different designs might be made on the basis of model tests in waves. In the past, the only reliable information on surfing ability has come from full scale testing and operational experience. Thus, the designer must rely heavily on past practice and experience with existing amphibians.

There are many characteristics which are directly related to the performance of an amphibian in surf. The range of stability indicates to what degree the amphibian can roll and yet return to an upright position. The static stability as determined by the metacentric height *GM* is a measure of the resistance of the amphibian to roll and is often given as a design requirement (unlike surfing ability, static stability is readily calculated; see par. 7-1.5.2). Maneuverability, especially in terms of quick and positive response to the helm, is related to the ability of this amphibian to operate in surf without broaching. Reserve buoyancy and provisions for automatic bailing of bilges and cargo wells are factors which affect the resistance of the amphibian to swamping. The speed of an amphibian may affect its ability in surf with respect to the ability to outrun waves (see par. 7-4.3.1).

A simple scaling law can be used as a general rule-of-thumb concerning surfing ability in the preliminary design stages. The maximum safe wave heights for similar amphibians are roughly proportional to their relative size. Length could be used as a measure of size, but for this purpose, the cube root of displaced volume, $\nabla^{1/3}$, is a better parameter. On this basis, an amphibian with twice the gross weight of another should be able to negotiate surf that is statistically 26 percent higher. As a rough estimate, the ratio of significant wave height $h_{1/3}$ (average of the 1/3 highest waves) in plunging surf to the cube root of volume can be taken as

$$\frac{h_{1/3}}{\nabla^{1/3}} \approx 1.2.$$

This corresponds to a wave height of 8 ft for the DUKW, which is considered as the safe limit for this amphibian. A SUPERDUCK was broached and turned over in plunging surf reported to be about 12 ft high at Monterey, California. A LARC V safely negotiated the same surf ($h_{1/3}/\nabla^{1/3} \approx 1.5$).

Another factor which should be considered for surfing ability is reliability. If the propulsion system or steering system should fail for any reason while the amphibian is in the surf zone, it is likely that the amphibian will broach, swamp, founder, and capsize. Similar consequences might follow a bilge pump breakdown. Self-bailing cargo wells are desirable for this reason.

The requirement for maneuverability is usually expressed in terms of a minimum tactical diameter, although minimum acceptable performance in some other maneuver may also be specified. The factors which determine maneuverability are discussed in par. 7-9.1. Simple model tests can be performed during the preliminary design stages in order to determine, qualitatively, the degree of directional stability of any candidate amphibian design. Scale effects may preclude accurate prediction of tactical diameter. The relative merits of different steering system configurations can be deduced from model tests, however. As with surfing ability, there are no convenient theoretical methods available to predict the maneuverability of proposed amphibian designs. The designer must rely on past practice, experience, or model tests. A simple scaling principle, however, can be applied to tactical diameter, i.e., the ratio of tactical diameter to length is roughly constant for similar amphibians. Of course, one must be careful in assessing similarity. For example, two amphibians of the same design could have grossly different turning ability, where one was directionally unstable and the other was made stable by virtue of an additional skeg or keelson, etc. A reasonable value of the ratio of tactical diameter to length (LOA) for amphibians with satisfactory maneuverability is 3.0. The subject of waterborne maneuverability is covered in more detail in pars. 7-9 through 7-9.4.

6-8 LAND PERFORMANCE

The land performance of wheeled vehicles is discussed at length in Refs. 6, 7 and pars. 7-8 through 7-8.5.2. The paragraphs which follow are included for general background.

6-8.1 SPEED AND TRACTIVE EFFORT

The maximum speed of an amphibian on land is a function of available power, rolling resistance, wind resistance, grade resistance, the suspension system, roughness of the terrain, and the type of soil. The gross tractive effort is the maximum propelling force that can be developed by the tires on a given type of surface. The limiting factor(s) determining the maximum speed is different for various environments. For example, over rough terrain and obstacles the speed is reduced for operator comfort and to avoid damage due to shock loads. On paved roads, power, especially on grades, may be the limiting factor. The natural frequency of the rigid suspension system is the limiting factor for the LARC V. In weak soils and sand, the gross tractive effort is limited by the strength of the soil and, at the same time, the rolling resistance of the tires is increased due to sinkage.

Rolling resistance is a function of the amphibian weight, tire design, speed, and the soil or other surface. Grade resistance is directly proportional to the weight and the slope of the grade. Wind resistance is a function of the frontal area and shape of the amphibian and its velocity, but is a small part of the total resistance at speeds below 35 mph. Where gross tractive effort is the limiting factor, the total resistance is just equal to the gross tractive effort at the maximum speed. This subject is covered in greater detail in par. 7-4.2.

6-8.2 SIDE SLOPE STABILITY

The side slope stability is a function of the vertical center of gravity and the suspension system design. With a rigid suspension system, the important parameters are the track width and the tire deflections. Fig. 6-9 illustrates the condition. The axial and radial tire deflections are caused by the increased loading on the downhill wheels. The radial deflection causes the amphibian to lean away from the slope so that the amphibian angle is greater than the slope angle. The axial deflection decreases the effective track width. Both effects are detrimental to the side slope stability. As the side slope increases the center of gravity moves toward a position over the center of pressure of the downhill tire. Once the center of gravity

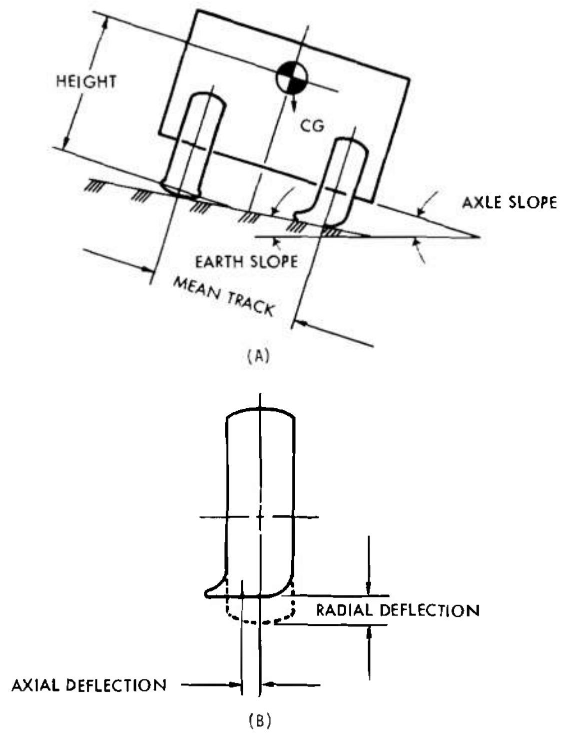


Figure 6-9. Side Slope Stability Definition Sketch

passes over this point, the amphibian will, of course, capsize laterally.

The limiting condition on side slopes is not necessarily the stability of the amphibian. For instance, the downhill tires on the LARC V would lose air and go flat on side slopes while the vehicle remained stable. This was caused by the axial deflection of the tire which pulled the sidewall away from rim, breaking the seal.

The side slope requirement is normally specified in the QMR.

6-8.3 APPROXIMATE MOBILITY INDEX

Mobility indexes provide a convenient quantitative measure of vehicle mobility. Some simple formulas for calculating the approximate mobility indexes, suitable for use in preliminary design, are given in Table 7-44. These indexes apply to soft soil and sand mobility.

6-9 MODEL TESTING

6-9.1 NECESSITY FOR MODEL TESTING

Model tests may be required at several stages of the design process in order to confirm

engineering estimates of diverse design variables or to obtain more accurate values of these variables when the estimates are rough or based on incomplete data. Model tests are needed especially for new and unconventional designs for which there is no previous experience or quantitative data which can be used to make estimates. While there are a number of successful operational amphibians in existence, they are certainly not considered to be optimum configurations with respect to waterborne performance. In any case, there does not exist a store of amphibian design experience or systematic series test data comparable to that which is available to the merchant ship designer.

Model tests are conducted in a controlled environment so that the effect of a single design variable can be found independently and the test results can be confirmed by simply repeating the test. Unknown variables such as wind and tide do not obscure the results. As long as there are no predominant scale factors involved, confidence in properly conducted model tests should be at a high level.

In summary, the gains which can be obtained by model tests are associated with the data which can be used to make design decisions with a high confidence level. These data can be obtained at relatively low cost and in a short time.

6-9.2 TYPES OF MODEL TESTS

The various types of model tests are described briefly in the paragraphs which follow. Greater detail concerning test procedures and modeling laws are found in pars. 7-4.1 to 7-4.1.3 and 7-7.6 to 7-7.6.4.

6-9.2.1 Resistance

The model is towed in a model basin over a straight path at various constant speeds. The drag, as a function of speed and displacement, is measured by a simple force gage or dynamometer. The drag component of appendages is deduced by running tests with and without appendages mounted on the model.

6-9.2.2 Self-propulsion

The model is propelled by a model propulsor in a model basin at the desired speeds. The rotative speed and power absorbed by the propulsor at each velocity are measured with suitable instrumentation. The test measurements are used, in conjunction with resistance and propulsor tests, to define the interaction between the hull and propulsor in terms of wake fraction and thrust deduction.

6-9.2.3 Maneuverability

Free running and captive model tests may be conducted to determine different aspects of maneuverability.

6-9.2.3.1 Free Running

The simplest test that can be made with a free running model is rarely used because it only indicates whether the amphibian is directionally stable or not. A self-propelled model with rudder fixed amidships is set on a straight course and then a momentary side force is applied at the bow. If the model establishes some new straight line path it is directionally stable, by definition. If it is directionally unstable, it will go into a steady turn after the force is applied to the bow.

Steady turning diameters can be obtained with the free running model by setting the rudder at fixed angle increments. However, more information can be obtained if the rudder is remotely controlled. Definitive maneuvers can be simulated with a remotely controlled free running model. The turning maneuver is described in par. 7-9.1.1 and Fig. 7-98. The Dieudonne spiral maneuver is described in par. 7-9.1.2 and Fig. 7-99. The spiral maneuver serves mainly to determine qualitative stability characteristics. One other maneuver, the Kempf overshoot (or zig-zag), serves to determine control characteristics. This maneuver is described in Ref. 7.

Free running model tests must be run in large turning or maneuvering basins. The model trajectory is recorded so that quantities such as yaw rate, turning radius, and speed may be determined. Model trim and heel in turns may also be observed.

6-9.2.3.2 Captive Model

The model is restrained by the towing carriage through a dynamometer in captive model tests

to obtain numerical values for the hydrodynamic coefficients in the equations of motion. With these coefficients, mathematical models, in conjunction with high-speed computers, are used to simulate the vehicle motions. This approach has the potential for obtaining more basic information on maneuverability than free running tests which only predict the motions but do not give insight into the controlling factors.

Three types of captive model tests are commonly performed: (1) steady state yaw, (2) planar motion mechanism, and (3) rotating arm. The first two are conducted in conventional towing tanks. The third must be run in a turning or maneuvering basin. The forces and moments on both the model and control surfaces are measured in these tests. A complete description of these tests, including how the results are used in the motion equations, is given in Ref. 8.

6-9.2.4 Seakeeping

Seakeeping tests are conducted in towing tanks equipped with wavemaking mechanisms. The model is generally towed into the waves (head seas) which may be either regular (constant length and height) or irregular, with long crests (waves coming from one direction only). If tests in quartering and/or short crested (waves coming from several directions) seas are desired, they must be performed in special seakeeping basins. The model may be free of pitch and heave or may be constrained in all six degrees of freedom. As with maneuvering tests, the free model tests are used to simulate the motions in waves and the captive model tests are used to measure forces and moments in order to derive the motion coefficients. These coefficients are used in the equations of motions from which the motions in arbitrary sea conditions can be calculated. These calculations require the use of a high-speed computer. This approach has the advantage that the motions in a wide range of sea conditions can be studied without the need to simulate each in a testing tank.

6-9.2.5 Propulsion

Propulsion tests are conducted to determine the propulsor performance coefficients (thrust, torque, and efficiency) over a wide range of operating conditions in open water. Tests may be performed either in a towing tank or a circulating water tunnel or channel. If cavitation effects are important, a water tunnel or channel with provisions for modeling cavitation number must be used. Thrust, torque, rotative speed, and advance speed are all measured. The data are usually reduced to the coefficient form as propeller characteristic curves. These tests are covered in detail in pars. 7-6 through 7-6.4.

6-9.3 SCHEDULE OF TESTS

Several factors should be considered when fixing a schedule for model tests:

a. Relation of model test to design process
b. Interrelation of tests
c. Lead time
d. Testing agency and facility

The model tests are arranged to provide data for decisions and estimates in the evolutionary design process. The test schedule should take into account the natural sequence of events in the design evolution. For example, the hull lines and gross vehicle weight must be known before a model test can be made to determine the resistance. This resistance, in turn, should be known before the propulsor characteristics are selected and propulsion tests are performed. Thus, the relation between tests such as resistance, propulsion, and self-propulsion tests must be considered.

The lead time for model tests includes time for model construction, facility availability, testing, and data reduction. These factors may depend greatly on the schedules and capabilities of the prospective test agencies and facilities, i.e, their work-load. The test schedule should be carefully considered when selecting a test facility because delays in obtaining test results will lead, indirectly, to increased costs in the design process.

Another very important factor which must be considered in selecting a suitable test facility is its size. Larger tank cross sections allow proportionally larger models to be tested without any increase in tank wall effects. The use of larger models is advantageous with respect to scale effects as explained in par. 7-4.1.1.3. In effect, the larger the model the more accurate the prediction of full scale performance will be.

6-9.4 POSSIBLE LOCATIONS FOR MODEL TESTS

Table 6-7 is presented to give a comprehensive list of test facilities which might be considered for amphibian model tests. Governmental, institutional, and private facilities are included. A brief description of the facility and any special capability is included for convenience. More complete information including foreign facilities is tabulated in Ref. 9.

TABLE 6-7 MODEL TEST FACILITIES

Name and Location	Type of Facility	Dimensions of Basin or Test Section (Length x Width x Depth)	Speed,* fps	Remarks
Aerojet-General Corp. Azusa, California	Ring Channel with Rotating Arm	Diameter = 110 ft Channel width = 20 ft Depth = 10 ft	[illegible]	
California Institute of Technology Pasadena, California	Variable Pressure Water Tunnel	14 in. Diameter, Free Surface 20 in. wide 20 in. deep	100 to 25	Pressure from atmospheric to 1/15 atmospheric used for body studies.
Collins Marine Laboratory Collins Radio Co. Cedar Rapids, Iowa	Towing Tank	90 x 6 x [illegible] ft	22	Has plate surface agitator.
Davidson Laboratory Stevens Inst. of Tech. Hoboken, N. J.	Towing Tanks (2)	313 x 12 x [illegible] ft 103 x 9 x 4.5 ft	100 21	Both tanks have plunger type wave-makers for uniform regular or irregular waves.
	Seakeeping Basin	75 x 75 x 4.5 ft	[illegible]	Wavemaker can generate regular or irregular waves and models can be run at all angles to the waves.
	Rotating Arm	75 x 75 x 4.5 ft	150	The rotating arm is in the sea-keeping basin above.
General Dynamics/Convair Hydrodynamics Labora-tories, San Diego, Calif.	Towing Tank	300 x 12 x [illegible] ft	100	Mechanical paddle type wavemaker.
HYDRONAUTICS, Incorporated Laurel, Maryland	Towing Tank	308 x 24 x 13 ft	25	Plunger type wavemaker for uniform regular or irregular waves.
	Variable Pressure Water Channel	12 x 2 x 2	40	Pressure from atmospheric to 3 in. Hg. Used for bodies, foils and propellers.
Iowa Inst. of Hydraulic Research, Univ. of Iowa, Iowa City, Iowa	Towing Tank	300 x 10 x 10 ft	20	
	Water Tunnel	13.5 in dia. or [illegible] x 24 in. cross-section	3-35	Pressure from 3 to 30 psia. Used for pressure distribution on bodies and foils.
Lockheed Underwater Missile Facility Sunnyvale, California	Towing Tank	180 x 15 x 15 ft	50	Controlled pressure from 0.73 to 14.7 psia.

TABLE 6-7 (CONTINUED)

Name and Location	Type of Facility	Dimensions of Basin or Test Section (Length x Width x Depth)	Speed, fps	Remarks
Massachusetts Inst. of Technology, Cambridge, Mass.	Towing Tank	108 x 8.6 x 40 ft	25	Hydraulic paddle wavemaker for uniform regular or irregular waves.
	Water Tunnel	24 in. diameter	34	Pressure from 0.8 to 14.7 psia. Used for propeller tests.
NASA Langley Field, Va.	Towing Tanks (2)	2800 x 24 x 12 ft	80	Flat plate wavemaker.
		2200 x 8 x 5 ft	200	High speed capability.
	Water Tunnel	2.5 in. diameter	43	Used for pressure measurements on foils and visual cavitation studies.
Naval Ship Research and Development Center (NSRDC-DTMB) Carderock, Maryland	Towing Tanks (4)	1886 x 51 x 22 ft	42	Pneumatic wavemaker.
		303 x 51 x 10 ft	42	Combined shallow water and turning basin.
		2968 x 21 x 10 ft	100	High speed basin.
		142 x 5.5 x 10 ft	10	Pneumatic wavemaker.
	Seakeeping Basin	360 x 240 x 20 ft,	24	Regular and irregular and short crested waves can be generated at any angle to the model.
	Water Channel	60 x 22 x 9 ft	17	
	Water Tunnels (2)	36 in. diameter	85	Pressure from 2 to 50 psia. Propellers and contra-rotating propellers.
		12 in. diameter	26	Pressure from 1 to 15 psia propellers.
	Rotating Arm Basin	260 x 260 x 21 ft	85	
Newport News Shipbuilding and Dry Dock Company Newport News, Virginia	Towing Tank	56 x 8 x 4 ft		
	Water Tunnel	7 x 24 in. cross-section	21	
Northwestern University Evanston, Ill.	Towing Tank	70 x 6 x — ft		Wavemaker
Ordnance Research Lab. Pennsylvania State College, Penn.	Water Tunnel	48 in. diameter	80	
St. Anthony Falls Hydraulic Laboratory, University of Minnesota Minnesota, Minn.	Towing Tank	250 x 9 x 6 ft	25	Hinged plate wavemaker
	Water Channels (3)	10 x 3 x 5 ft	5	Low speed channel
		50 x 1.5 x 1.0 ft	45	High speed channel
		40 x 2.5 x 3.33	10	
	Water Tunnel	6 in. diameter	8-54	Pressure from 1 to 14.7 psia

TABLE 6-7 (CONCLUDED)

Name and Location	Type of Facility	Dimensions of Basin or Test Section (Length x Width x Depth)	Speed, fps	Remarks
University of Calif. Berkeley, California	Towing Tank	200 x 8 x 6 ft	6	Hydraulic paddle wavemaker can produce uniform regular or irregular waves.
University of Michigan Ann Arbor, Michigan	Towing Tank	360 x 21 x 9.5 ft	20	Plunger type wavemaker for uniform regular waves.
U. S. Naval Academy Annapolis, Maryland	Towing Tank	86 x 6 x 4 ft	20	Pneumatic wavemaker.
Webb Institute Glen Cove, N. Y.	Towing Tank	73 x 10 x 5 ft	16	Plunger type wavemaker for uniform regular or irregular waves.

CHAPTER 7 DETAILED DESIGN

SECTION I HULL DESIGN

7-1 HULL LINES

A fundamental part of the hull design process is the development of the hull shape or lines. This part deals with the drawing of the hull lines, some of the considerations involved in selecting the shape, and the calculations required for support of the drawing process.

7-1.1 GENERAL DESCRIPTION AND EXPLANATION

The hull shape of an amphibious vehicle is rather complex so that a systematic method of describing it must be employed. To accurately delineate the hull shape, three sets of planes are used to slice the craft along three orthogonal axes. The three sets of planes are the buttock, section, and waterline planes.

The *buttock planes* cut the hull along its length and are parallel to a vertical plane.

The *section planes* cut the hull into transverse vertical slices. The section planes are rotated 90 deg from the buttock planes.

The last of the orthogonal set of planes are the *waterline planes* which cut the craft into horizontal slices, parallel with the baseline plane.

The intersection of the hull surface with each section plane is described by an outline on the cutting plane.

The composite drawing of the outlines from each successive cut of the section planes is called the body plan. Similarly, the half-breadth and profile plans are composite drawings of the outlines obtained by intersection of the hull and waterline planes and buttock planes, respectively.

The three drawings—body, half-breadth, and profile plans—are collectively called the lines drawing or often just the "lines", Fig. 7-1 is a lines drawing of the LARC V, showing the three plans.

In addition to the body, profile (or sheer), the half-breadth plan, it is customary for the lines drawing to contain a table of offsets. The offsets are the dimensions to the outside of the shell plating. These dimensions are given at each station for the distance of the intersection of each buttock plane with the shell above the base line and the distance of the intersection of each waterline plane with the shell from the centerline. The dimensions in the table of offsets in Fig. 7-1 are given in inches and tenths of inches. It is customary in naval architecture to give the offsets in feet, inches, and eighths of an inch and to give offsets to the inside of shell plating. This type of offset table may be encountered in the lines drawing of some wheeled amphibians.

It is customary in a lines drawing of a wheeled amphibian to label the section, buttock, and waterline planes with a dimension which is the distance from an appropriate reference plane. The reference plane for the section planes may be established at the extreme forward point (or aft point) of the hull. The reference plane for the buttock planes is the centerline plane. The reference plane for the waterline planes is an arbitrary baseline plane. In general the baseline plane is established on the hull bottom or keel.

7-1.1.1 Lines Fairing Procedure

The basic steps in drawing a set of lines are given in par. 6-4.5.3. These steps apply to the final lines as well as the preliminary lines and need not be repeated. The major difference is that the final lines are done to a larger scale and contain more section, buttock, and waterline planes for improved accuracy.

For the reasons given in pars. 7-1.2 through 7-1.2.4, the waterborne performance of a wheeled amphibian will be improved if the hull lines are fair. A fair line is one which is smooth and continuous, i.e., without rapid changes in the radius of curvature. A line which is fair in three different orthogonal projections will be fair in all projections. This is one of the reasons that three orthogonal sets of planes are used to describe the hull shape.

In developing a fair set of lines, the key is to produce lines that are geometrically correct as

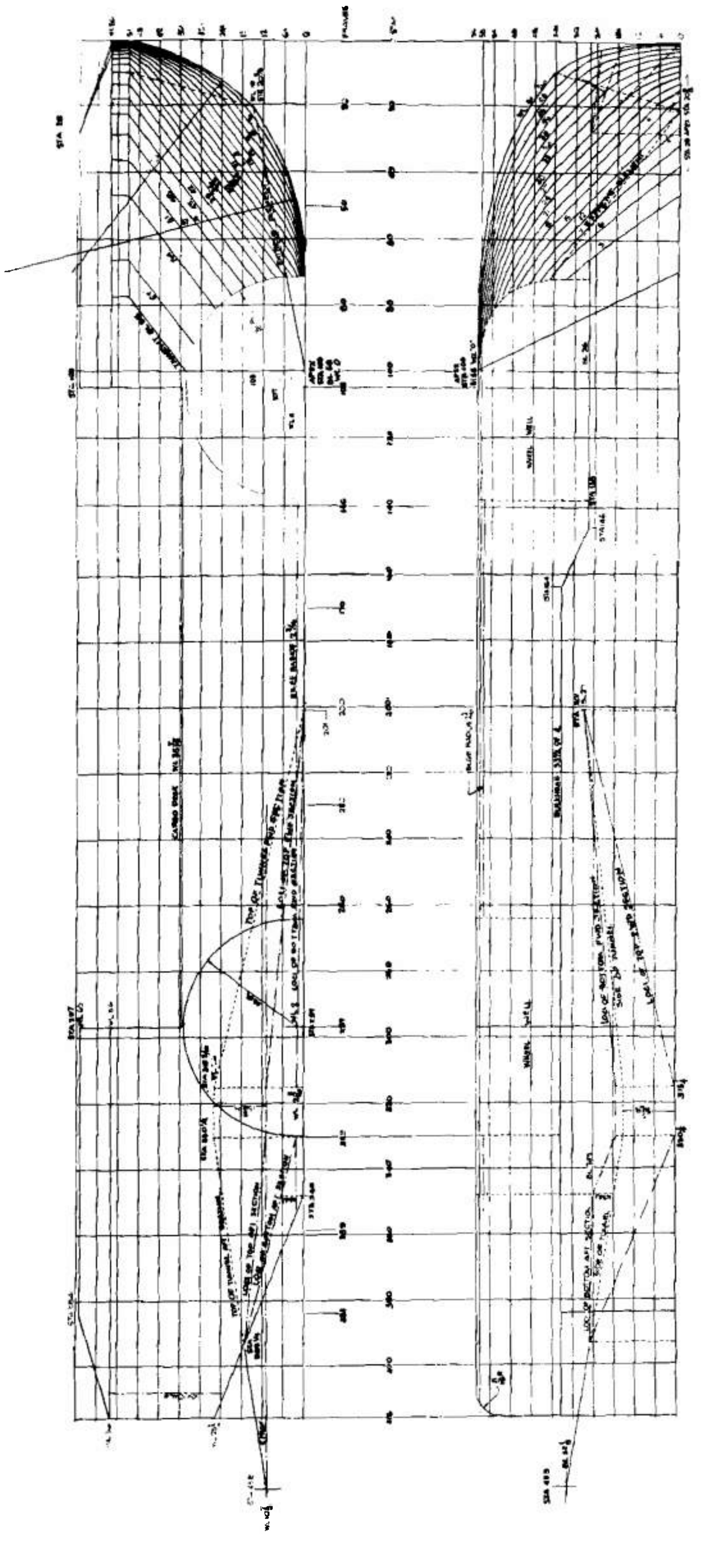
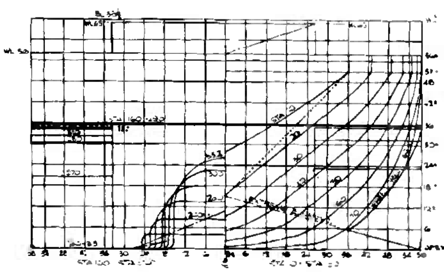
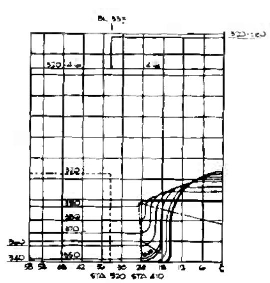


Figure 7-1. LARC V Lines Drawing (1 of 2)





BUTTOCK HEIGHTS AT STATIONS																			
STA	1	3	6	9	12	15	18	21	24	27	30	33	36	39	42	45	48	51	54
0	51.0																		
10	27.7	29.3	30.7	32.4	34.2	36.0	38.0	39.7	42.0	44.1	46.4	48.6	51.0						
20	15.6	18.0	20.3	22.5	25.0	27.2	29.6	32.0	34.3	36.5	39.0	41.2	43.6	46.4	49.5				
30	9.5	11.1	13.0	15.3	17.6	20.0	22.3	24.6	27.0	29.3	31.6	34.0	36.3	39.0	42.1	45.6	50.0		
40	5.2	5.7	6.7	8.3	10.3	12.5	15.0	17.3	19.6	22.0	24.3	26.6	29.1	31.7	35.0	38.5	42.6	48.0	
50	2.3	2.5	3.0	3.6	4.5	6.0	7.7	10.0	12.3	14.6	17.1	19.3	21.7	24.4	27.5	31.0	35.3	40.6	48.4
60	0.5	0.6	0.7	1.1	1.4	2.0	2.5	3.5	5.2	7.4	9.7	12.1	14.4	17.2	20.4	24.0	28.1	33.2	41.1
70	0.0	0.0	0.1	0.1	0.2	0.3	0.5	1.0	1.3	2.0	3.0	4.7	7.3	10.1	13.2	16.6	21.0	26.1	34.0
80																			26.2
90																			
100																			

HALF BREADTH AT WATERLINE																			
STA	3	6	9	12	15	18	21	24	27	30	33	36	39	42	45	48	51	54	56
0																			
10										4.2	9.6	14.4	19.5	23.7	28.1	32.0	36.0	36.0	36.0
20						3.2	6.6	10.5	14.5	18.3	22.2	26.0	30.0	33.7	37.3	40.4	43.3	43.3	43.3
30				4.3	8.4	12.3	16.1	20.1	24.0	27.6	31.5	35.4	39.0	41.7	44.3	46.5	48.5	48.5	48.5
40		3.2	9.7	14.1	18.0	21.7	25.5	29.4	33.2	37.0	40.1	42.7	45.3	47.4	49.3	51.0	52.3	52.3	52.3
50	5.4	15.1	19.5	23.4	27.3	31.2	35.0	38.4	41.5	44.2	46.4	48.3	50.1	51.4	52.6	53.7	54.7	54.7	54.7
60	19.1	25.1	28.7	32.6	36.4	39.6	42.4	45.0	47.2	49.1	50.6	52.1	53.2	54.2	55.0	55.6	56.4	56.4	56.4
70	29.7	34.3	37.6	40.6	43.4	45.7	48.0	49.6	51.3	52.5	53.5	54.4	55.2	55.7	56.4	57.0	57.5	57.5	57.5
80								53.1	54.2	55.0	55.5	56.0	56.5	57.1	57.4	57.6	58.0	58.0	58.0
90																			
100																			

SHOWN IN INCHES

Figure 7-1. LARC V Lines Drawing (2 of 2)

well as fair. Geometrically correct means that all three views (the body, profile, and half-breadth planes) must represent the same hull surface. This requires that all intersection points among the three sets of orthogonal planes be consistent in all three views on the lines plan. For example, the distances from the centerline plane to the waterlines in the half-breadth plan measured at each section plane must be the same as the distances from the centerline plane to the sections in the body plan as measured at each waterline plane. Similar relationships must exist between the distances of the buttocks above the baseline plane in the profile plan and body plan, and between the distances of the intersections of waterlines and buttocks from the section planes in the half-breadth and profile plans.

It must be remembered that when a line is changed in any of the three views on the lines plan it must be corrected in the other views so that the geometric accuracy is maintained. Thus the development of a set of lines that is fair and geometrically accurate is an iterative procedure which requires many cycles. An easy way of rapidly checking the geometric accuracy between views is to mark a strip of paper with a tick mark at an intersection and at a reference plane in one view and then compare the marked distance on the paper strip with the distance from the reference plane to the intersection in another view. This takes less time and is more accurate than measuring the distances with a scale.

7-1.1.2 Developable Surfaces

7-1.1.2.1 Advantage of Developable Surfaces

If the hull is to be of metal, it is desirable that the final lines be developable. A developable surface is one which can be represented by the surface of a cone with a movable or fixed apex. A cylindrical surface is also developable. Plating for developable surfaces can be formed by relatively simple rolling techniques or cold bending in place over frames or jigs.

Accordingly, developable surfaces are used in metal hulls for simplicity of construction with resulting lower cost. Special shaping techniques necessary for more complex shapes are very costly, time-consuming, and require special equipment. Further, vehicles with developable surfaces can be repaired in shops which have minimal equipment.

7-1.1.2.2 Amphibian Hull Developable Surfaces

There are, essentially, only three surfaces on an amphibian hull which depart from the box-shape—the spoon-shaped bow which is the largest curved hull surface, the propulsor tunnel, and the corners.

The corners at the intersection of two flat plates can be simple circular arcs. The tunnel can be shaped as a combination of cylindrical and conical surfaces.

The bow of a wheeled amphibian can be made up of a combination of developable surfaces. The bow of the LARC V, illustrated in the profile and half-breadth plans of Fig. 7-1, is an example of this. The bow shape of the LARC V consists of four different developable surfaces—two cones, a flat, and a cylinder. The cones are located between the dotted lines (one is labeled EXTREME ELEMENT) and the centerline keel. The flat surface is located between the dotted lines, and buttock 39 in the profile plan. The cylindrical surface starts in the vicinity of buttock 39 and extends up to the sheer line (or deck at side).

In developing the bow surface, the designer is advised to begin with arbitrary elements similar to those in Fig. 7-1. Each of the surfaces—the two conical and one cylindrical—are developed independently. For example, when the two elements and other boundary for the flat are chosen, the three surfaces can be developed independently. Once the surfaces have been developed, body plans for the bow are obtained and compared with the body plan from the preliminary lines process. If the body plans do not agree, new elements and a new flat surface are chosen, the three surfaces developed again, and the resulting body plans compared. This is, quite obviously, a trial and error process.

7-1.1.2.3 Surface Development Procedures*

The first step is to draw the profile and half-breadth view of the bow and part of the parallel section through the cargo space. The waterline scale, which is the vertical scale in the profile plan, should be greater than the station and buttock scales. The purpose of this deliberate distortion of the profile plan serves to increase the curvature of the keel (and sheer line, in the case of vehicles with a boat-like sheer

* The major portion of this paragraph is reproduced from a paper by Kilgore, Ref. 1.

line), thus making tangents easier to obtain. A similar distortion in the half-breadth plan may also be necessary.

A simple surface development scheme is described and three methods are outlined. The designer will find the first and third methods most often used for the amphibian bow development. The lines of a small boat are developed as an illustrative example.

The important properties of a developable surface are:

1. A developable surface may be depicted graphically by drawing the family of planes that are tangential to it.

2. Any tangential plane to the surface is also tangential to any curve lying in the surface and intersecting the line of tangency, or ruling.

3. Any plane found to be tangential to the surface at any point is unique.

These properties, as far as drawing is concerned, suggest the following constructional approaches:

1. Planes tangential to the surface should be found.

2. These planes are best found by using given curves already known to lie in the desired surface, such as chine, sheer, and profile centerline outline.

3. If any plane can be found to be tangential at a particular point, it is the unique plane tangential at that point and may be used with confidence.

The best means of illustration is a practical example. The example which follows gives the general procedure for the development of planes applicable to a lines drawing of a vessel with the following particulars:

Length (waterline) LWL = 39.5 ft
Displacement Δ = 15.57 tons
Longitudinal center of buoyancy
LCB = 0.53 LWL abaft Station 0

A preliminary hull design drawing, meeting these requirements with straight line sections, is shown in Fig. 7-2. Sheet materials can hardly ever be applied to a hull of straight line sections, hence these sections must become suitably curved. The designer, with practice, will learn how to make allowances for this curvature in the preliminary design stage, so as to finally yield the required hull parameters.

The sides of the boat above the chine are not as easy to develop as the bottom below it. In Fig. 7-3 the hull drawing has been laid out, foreshortened longitudinally, in half-breadth and profile. This is done because the main method of constructional drawing consists of locating various points by drawing tangents to curves at these points.

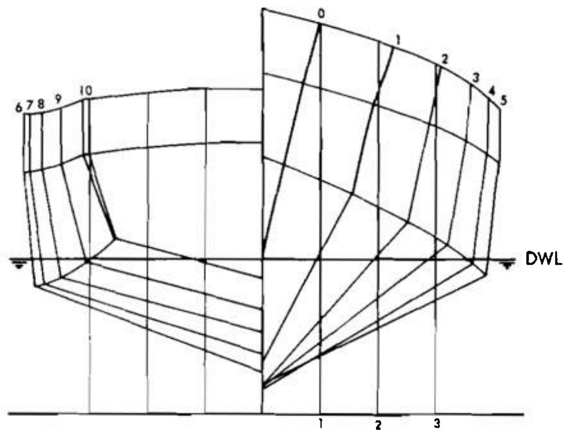


Figure 7-2. Preliminary Body Plan

The side surface of the boat above the chine is obviously contained by two required curves, the sheer and chine lines, but some doubt exists at present whether a developable surface can be fitted to the straight stem line. As the sheer and chine lines are not simple curves, we cannot develop the surface by the simpler method that will be utilized on the bottom. The approach used is to draw a tangent to one curve and the plane generated is then swung until it comes into tangency with the other known curve. This is not difficult, as is shown by the following procedure:

1. Take a point P at the bow on the chine and, in both the half-breadth and profile, draw the tangents to the chine line passing through P.

2. From this tangent draw any line AB intersecting the sheer line ,in B at a reasonable intersection angle in profile and half-breadth view.

3. At suitable short intervals from this line, i.e., at C and E, draw the same lines parallel to AB in the profile and the half-breadth.

4. At the intersection of the lines through C and E, and the sheer line in the half-breadth plan are the points D and F.

5. Transfer these points to the profile giving curve BDF.

6. Define the point Q where the curve BDF cuts the sheer line.

7. Take the point P' at the midpoint of BQ.

8. Join P and P' giving the line of tangency

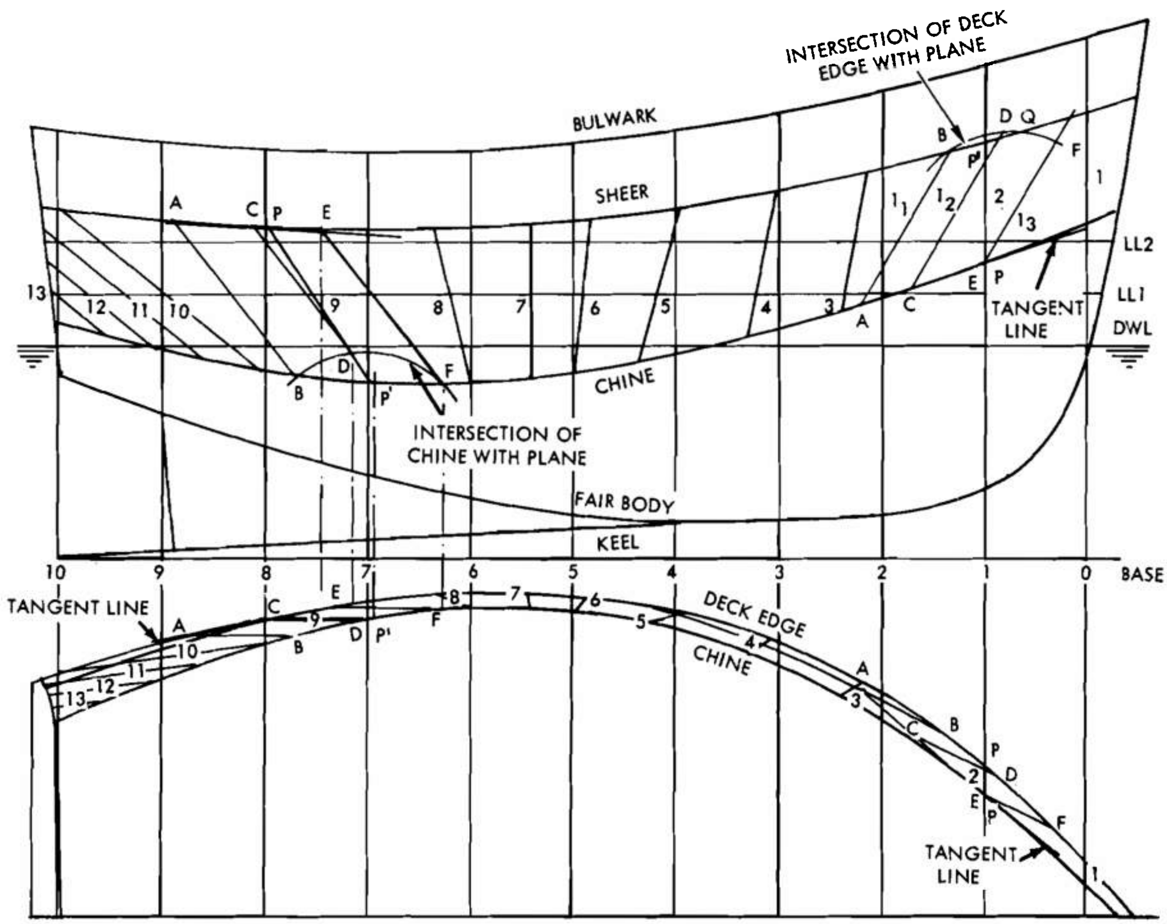


Figure 7-3. Development of Sides

(ruling) of the plane *PAB* with the side surface of the vessel.

An explanation of the above procedure follows: The line *ACE* is the tangent to the chine line at *P* in both views. Therefore, *PAB* gives a plane that is tangential to the chine and cuts the sheer line. If in profile this plane cuts the sheet line again at a second point adjacent to *B*, the plane must be almost tangential to the sheer line. To find this second point, we suppose the sheer line in the half-breadth to be the edge view of a cylinder. This cylinder cuts the plane *PAB* and because *AB*, *CD*, and *EF* lie in the plane *PAB*, the cylinder must cut the lines at the points *B*, *D*, and *F* and these lie, as already stated, on the sheer line in half-breadth. Therefore, the curve *BDF* in profile shows the intersection of the cylinder with plane *PAB* and coincides with the sheer line at *B* and the second point *Q*.

Now, if *PAB* is rotated about *PA*, it would become completely tangential to the sheer line approximately at the midpoint of *BC* at *P'* (osculating circle). The line *PP'* is, therefore, a line of tangency of a plane with the surface of the boat.

Another similar operation is illustrated farther aft. This time the initial tangent line is taken at the deck edge rather than on the chine. The sequence is the same as before: beginning at *P*, a tangent line to *A*; from *A*, on the tangent line, to *B* on the chine; parallel lines *CD* and *EF* to the chine in half-breadth plane. In profile, the lines are drawn initially from *C* and *E* parallel to *AB* without knowing where *D* and *F* occur on them. The locations of *D* and *F* in the profile view must be located by projecting them upwards from the lines in the half-breadth view. The curve *BDF* is the intersection of a cylindrical surface (of which the chine in this

case is the edge view in half-breadth view. The curve *BDF* is the intersection of a cylindrical surface (of which the chine in this case is the edge view in half-breadth) with plane *PAB*. This curve intersects the chine at *B* and *F*, and therefore a slight rotation of *PAB* will bring it into tangency about midway between *B* and *F*, at *P'*. The line *PP'* is a line of tangency of a plane with the surface of the boat, shown as ruling number 9, Fig. 7-3.

The rest of the rulings on the side surface, shown in heavy lines and numbered 1 through 13, were obtained by the same procedure. The procedure will require approximately two hours. It should be noted that development of the planes was commenced in the half-breadth view. It could have been started in the profile view, letting the sheer or chine in that view represent the edge view of the cylinder, but then the cylinder would have intersected the plane almost perpendicularly, and the curve of intersection when projected on half-breadth would not be so well defined. With experience, the draftsman will initially select the view where the plane will be intersected at the most acute angle. Occasionally the two points of intersection of the plane with the given line will be too far apart, in which case it will be necessary to refine the accuracy by drawing a new plane closer to the point of tangency.

Development of the bottom is much simpler because the centerline curve lies entirely within a plane. Because of the foreshortening of the drawing, the resultant curvature of the lines makes points of tangency easier to find. The development of the surface near the bow is shown in Fig. 7-4. Ruling 2 is used as the example. The graphical construction follows:

1. In half-breadth draw tangent to chine line at *P* and project it forward to intersect with the centerline plane at *A*.
2. Transfer the point *P* to the profile and draw the tangent to the profile chine line through *P*.
3. Transfer point *A* from half-breadth to extended tangent line through *P* in profile giving the point *B*.
4. Through *B* draw tangent line to bow profile obtaining point *C*, giving the plane *PBC*.

The explanation of the above procedure follows: The intersection of the tangent with the centerline at *A* in the half-breadth must be in the same plane as the point *B* in profile. The line from *B* locates *C* and is tangent to the stem profile. Hence the tangential plane is complete in *PBC* because it is a plane tangential to two curves lying in a developable surface. Line *PC* is the unique line of tangency of *PBC* with the surface. Hence *PC* is ruling number 2. Rulings numbered 1, 3, and 4 are determined by the same method.

Beyond Station 4, a tangent to the chine will run off the drawing-board before it intersects the centerline, so that from this point a third method of development must be employed. The method used follows (Fig. 7-5):

1. Draw a line parallel with the centerline tangential to the chine line giving the point *P* in the half-breadth.
2. Transfer this point to the profile and draw the tangent to the chine line through *P*.
3. Still in profile, draw a line parallel to this tangent line to the chine, but tangential to the centerline outline giving the point *A*.
4. Join *PA* which is the ruling of the surface number 5.

This procedure is based on the theorem of parallel tangents already stated. Therefore, by the theorem, the tangent to the chine at a ruling of the surface through *P*, in the profile view, will be parallel to the centerline outline curve at the intersection of that ruling with the centerline outline.

Now referring to Fig. 7-2, note that bottom sections 8, 9, and 10 are parallel in the body plan. They are also parallel in all other views, and, therefore they define a cylindrical surface. Hence we do not have to develop the bottom abaft Station 8; this is already done.

To obtain the rulings 6 to 9, the following procedure is adopted:

1. Transfer the buttock lines 1, 2, and 3 to profile by using the height intersection from the body sections for sections 8, 9, and 10. Transfer the intersections of the buttocks and rulings 1, 2, 3, 4, and 5 in the half-breadth to the profile. Fit a spline to these points precisely, and draw the buttock lines in profile.
2. Draw parallel tangents to the buttocks and centerline profile line in profile; the points of tangency give the rulings.
3. Check that all points lie on straight lines in both views. This gives the points *BB'*, etc., and the rulings 6, 7, and 8.

This again is dependent on the parallel tangent theorem as buttocks and centerline

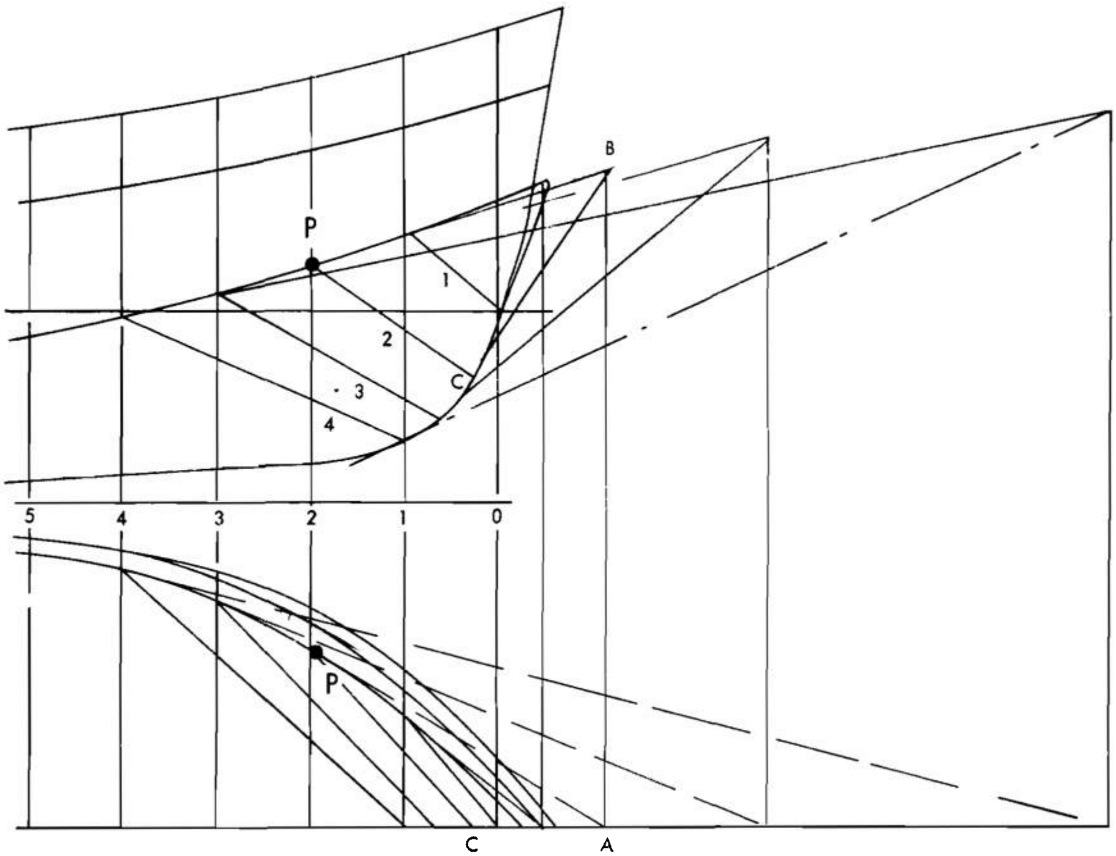


Figure 7-4. Development of Bottom by Intersecting Tangents

profile represent parallel planes. The gap between Station 8 and ruling number 5 is considerable, but the spline is held on at least three points at each side of the gap. It will fairly accurately describe the only fair line that can be drawn between these points. The balance of the bottom, abaft Station 8, of course, has already been found to consist of straight, parallel frames.

Developing the bottom should require a half hour. If it had been necessary, the same method could have been used on the bottom as on the sides, but faster and more accurate methods were available. The final lines are shown in the body plan, Fig. 7-6. This was obtained by transferring the new buttock lines on the stations from the profile and half-breadth shown in Fig. 7-5. The sections in the sides of the forward part of the boat are so nearly straight that the very slight curvature in them cannot be shown with the small scale on which these drawings were done. Almost any material, metal or otherwise, will have sufficient plasticity to accommodate slight deformation. The development of the bulwarks is not shown. In this case the top of the rail is drawn at a constant distance from the deck, and the bulwarks could be rapidly and accurately developed by the method of parallel tangents because over short distances the sheer and bulwark lines in profile can be considered straight lines.

The displacement, and the position of the longitudinal center of buoyancy have been altered, although not a great deal. If the exact specified particulars are desired, the designer should make provision in the preliminary plan for these predictable changes. The lines of chine, sheer, and outline are exactly the same as originally proposed.

7-1.2 WATERBORNE SPEED AND RESISTANCE CONSIDERATIONS

At the point in the design process that the

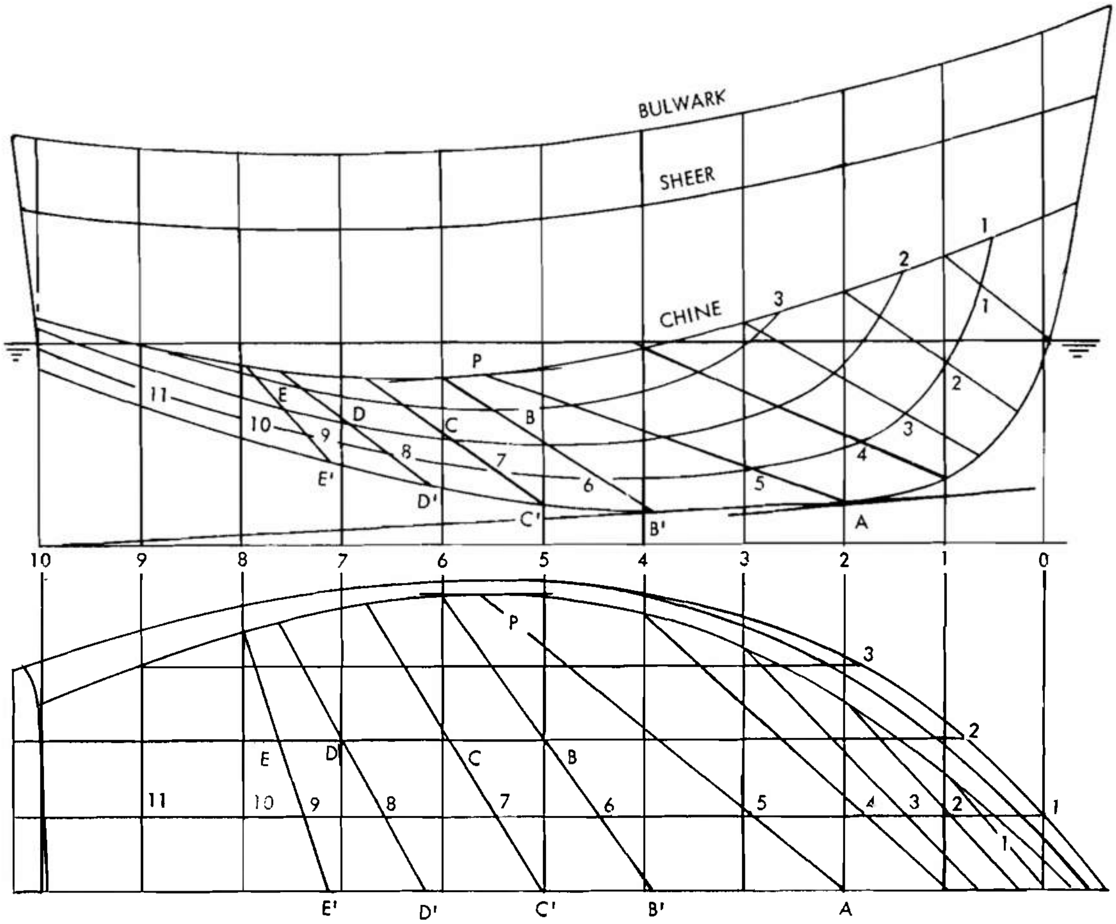


Figure 7-5. Use of Parallel Tangents

hull lines are drawn, the basic hull form requirements—such as dimensions, displacement, center of buoyancy, and general shape—will have been established. Within these constraints the details of the hull shape must be developed. The actual details of the shape depend on both hydrodynamic and hydrostatic considerations. The remainder of this paragraph on hull lines deals with the relationships between hydrodynamic and hydrostatic considerations and the details of the hull shape. It must be realized that many of these considerations conflict with one another so that compromises must be made on the basis of the designer's judgment and in light of the overall design objectives.

7-1.2.1 Hull Shape and Fairness

The basic area of the hull in which the designer has a range of shapes to choose from is at the bow. A review of Fig. 7-1, the figures in Chapter 3 showing conventional amphibian hulls, and the figures in Chapter 9 showing high-speed amphibian hulls indicates the variations possible. These can be divided into two basic types, i.e., the scow bow, as exemplified by the DUKW, and the faired or boat-type bow as exemplified by the LARC V. Par. 7-4.1.2.2 and Fig. 7-61 indicate the effect of bow form on the calm water resistance. It has been concluded from calm water model test data that the effect on calm water resistance is not large. It is expected that the bow shape will have a larger effect on rough water resistance and seakindliness. In these areas the faired bow should be superior to the scow shaped bow. However, there are no quantitative data available to evaluate the effects or suggest details of the shape. Experience with small boats indicates that for seakindliness, the bow should be given

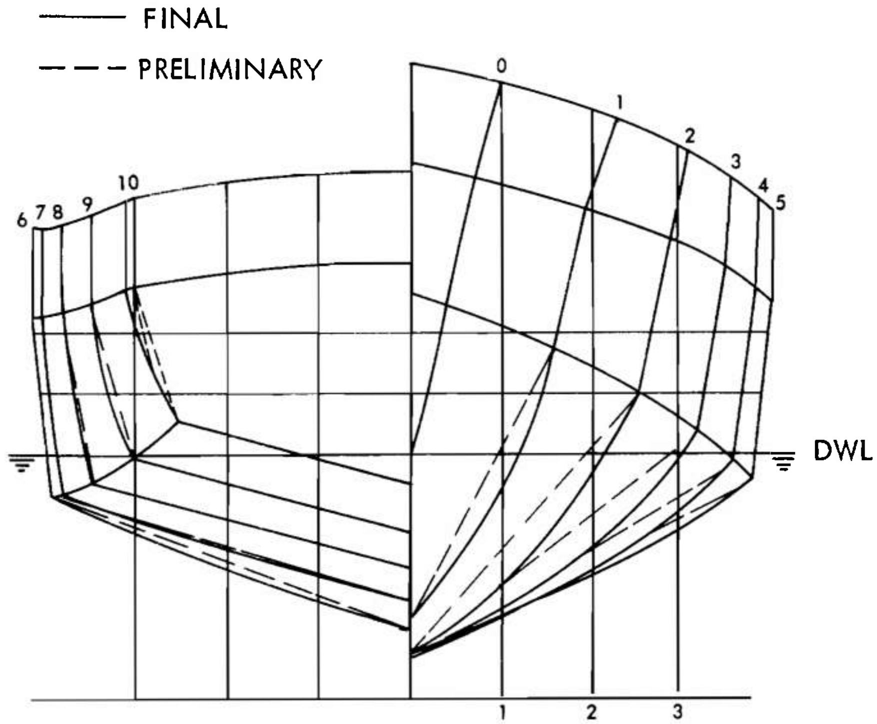


Figure 7-6. Final Body Plan

the maximum possible freeboard and flair and the minimum possible entrance angle. It is the opinion of one of the LARC V designers that the prototype LARC V, which had more flair and a smaller entrance angle than the production LARC V, shipped less water in head seas and was, generally, more seakindly.

The concept of a fair surface is presented in par. 7-1.1. By the absence of rapid changes in radius of curvature, the fair surface reduces the change of local flow separation or turbulence which results in the formation of eddies and increases the eddymaking component of the residuary resistance (see par. 7-4). Thus, it is desirable for the designer to provide a fair shape where possible.

7-1.2.2 Chines and Spray Strips

The requirements for internal volume and, in most cases, developable hull surfaces require that an amphibian hull be designed with chines. A chine can be defined as an intersection between two fair surfaces. Fig. 7-3 illustrates a typical chine on a small boat. Across a chine the hull is, of course, unfair and, as a result, flow across the chine may cause local eddymaking and increased resistance. Therefore, if possible, chines should be aligned parallel to the direction of the local flow. In many cases this will not be possible. In this event a sharp chine should be replaced with as large a radius as possible. The intersection of the hull side and bottom amidships in the LARC V lines shown in Fig. 7-1 is an example of such a condition. An exception to the idea of rounded chines occurs in the case of high-speed amphibians where a sharp chine will cause the flow to break away which reduces wetted surface.

A spray strip is often fitted to high-speed amphibians to "peel" the spray off the hull surface, reducing frictional drag. The spray strip can have a triangular or square cross section with sharp edges. A typical spray strip cross section and location are illustrated in Fig. 9-9. In practice the location of the spray strips is often determined by trial and error in model tests.

7-1.2.3 Wheels and Wheel Wells

As indicated in par. 7-4, the exposed wheels of a conventional amphibian are a major source

of drag. This drag is basically a function of exposed frontal area so that little can be done to reduce it by altering hull shape. However, if possible, exposed projections on wheel rims and wheel ends should be faired or recessed into the hull. From considerations of resistance and stability, the wheel wells should be kept as small as possible. The minimum width is determined by clearance requirements for steering. Model tests of the LARC V hull arranged for 2- or 4-wheel steering with the same turning radius showed little difference in resistance. Radial clearance between the tire and wheel well is required to keep loose objects from being wedged between them. Experience with the LARC V indicates that it has been provided with about the minimum acceptable radial clearance.

7-1.2.4 Appendages

The major appendages on a wheeled amphibian are associated with the propulsion, steering, and suspension systems. The design of the appendages for the propulsion and steering systems are covered in pars. 7-7 and 7-9. The appendages involved with the suspension system can have a significant effect on the details of the hull shape. These appendages include items such as axles, wheel ends, and differential housings. In an amphibian with a rigid suspension, the only exposed item of this type will be the wheel ends and associated gear box. These should be recessed into the hull as far as possible or appropriately faired when exposed. In the case of an amphibian with a spring suspension system; exposed axles, differentials, or other appendages should be housed in hull recesses. This was done on the DUKW and found to significantly reduce the total resistance.

7-1.3 WATERBORNE PROPULSION CONSIDERATIONS

It is probable that any wheeled amphibian design propelled with a screw propeller will require a propeller tunnel. The function of the propeller tunnel is to provide inflow to a propeller that has been recessed into the hull for protection and to increase clearance on land. In conventional amphibians of the LARC series the propeller operates in the tunnel at all times; whereas in high-speed amphibians the propeller only operates in a tunnel during surf operations. In either case the highest possible propulsion efficiency is required, hence care is required in designing the tunnel.

From the standpoint of propulsion, the inlet section of the tunnel (i.e., from the forward end to the propeller) is most critical. For highest propulsive efficiency and minimum vibrations, it is desirable that the inflow to the propeller have a uniform velocity. It is important, therefore, that flow separation does not occur in the tunnel inlet section. In this respect the tunnel design problem is similar to that of diffuser, condenser scoop, water jet inlet, or air scoop design. Refs. 2 through 9 provide design information on this subject.

It is of value to consider experience with the LARC series tunnel designs since the propeller tunnel is not exactly analogous to the items covered in the references. Data on the LARC series tunnels are presented in Fig. 7-7. The most critical part of the tunnel inlet geometry is the inlet angle. If this angle is too large, flow separation will occur. The values of this angle in the LARC series ranged between 10.5 and 13.5 deg.

Ref. 10 indicates that for flush waterjet inlets the maximum angle should not exceed 15 deg. Every effort should be made in the tunnel design to keep the inlet angles below this value. If there is some compelling reason to increase the inlet angle, the design should be model tested to determine if separation exists and the form modified if necessary.

The LARC LX was the first of the series designed and was found to have significant propeller vibration. To solve this problem on the later LARC V and LARC XV designs, the sides of the inlet were opened transversely. This had the effect of increasing the tunnel volume and altering the tunnel inlet sectional area curve, as shown in Fig. 7-7. It is reported that the propeller induced vibrations of the LARC V and LARC XV are less than those of the LARC LX. Further information on tunnel design and the effects of propulsion requirements on hull shape is presented in par. 7-7.

7-1.4 WATERBORNE MANEUVERABILITY CONSIDERATIONS

The requirements for adequate maneuverability will have important effects on the hull shape in the area of the maneuvering device. The basic concepts and types of

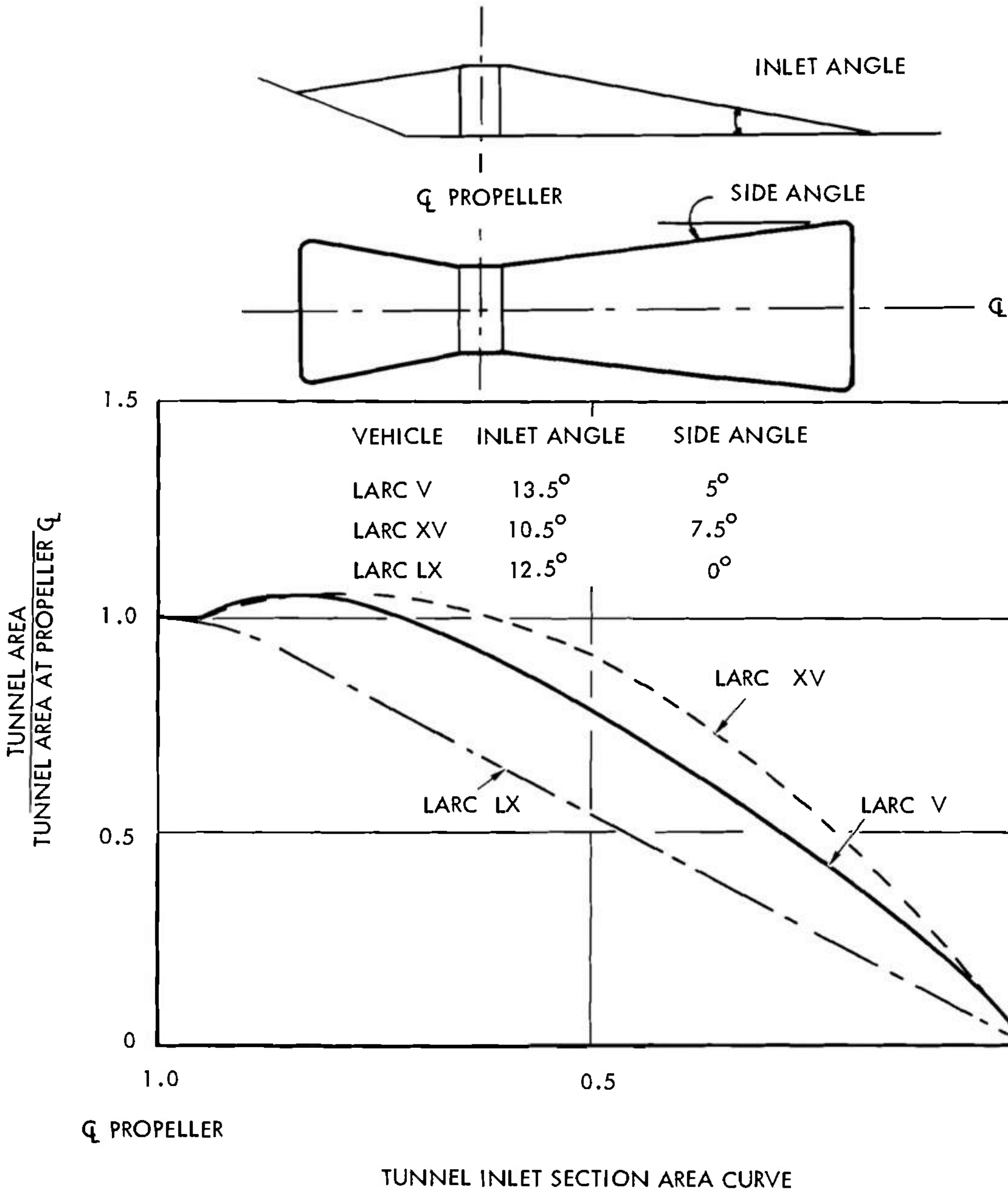


Figure 7-7. Wheeled Amphibian Tunnel Data

maneuvering devices are covered in detail in par. 7-9. In order to operate effectively; rudders, right angle propeller drives, and vertical axis propellers must have unrestricted inflow and outflow. In the case of a passive maneuvering device such as a rudder, adequate inflow is provided even at low speeds by placement in the propeller slipstream. Active maneuvering devices which rotate the propulsion thrust vector must be provided inflow according to the considerations covered in pars. 7-1.3 and 7-7.

Fig. 7-8 shows the interaction between rudder and tunnel. Outflow from the maneuvering device must be provided over the range of angles through which the slipstream is deflected. Any projection in the deflected slipstream may be

expected to produce drag forces which will act against the turning forces. For this reason the sides of the propeller tunnel, aft of the propeller, are flared outward in all amphibian designs. The angle of the tunnel side with respect to the centerline ranges from 12 deg on the older LARC LX to 14 deg and 16 deg on the newer LARC V and LARC XV, respectively. The depth of the tunnel is rapidly reduced aft of the propeller on the LARC series by the cutaway of the hull required to produce an acceptable approach or departure angle.

The importance of opening up the propeller tunnel aft of the propeller is illustrated by the model tests of the LARC V reported in Ref. 11. A configuration was tested in which the tunnel sides were extended straight aft from the propeller by means of flaps. This had the effect of reducing the side force for maneuvers to 20 to 40 percent of that developed with diverging tunnel sides.

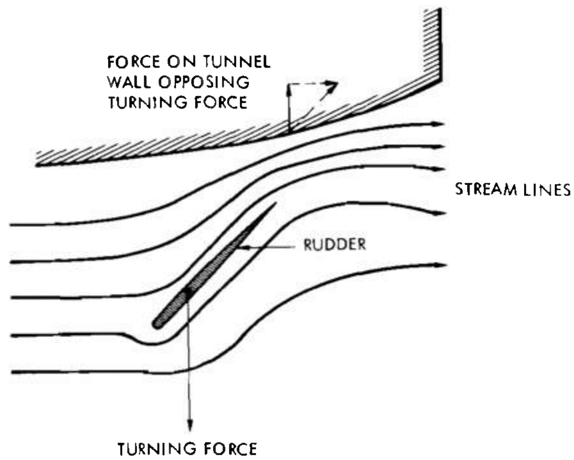


Figure 7-8. Interaction Between Rudder and Tunnel Walls

7-1.5 SEAKEEPING AND SURFING CONSIDERATIONS

To operate successfully in waves and surf, a wheeled amphibian must have adequate power, control or maneuverability, stability, and watertight integrity. These areas and their effect on hull shape are covered in pars. 7-1.4, 7-1.6, 7-4.1.2.4, 7-4.3.1, and 7-9.

7-1.6 BUOYANCY AND WATERBORNE STABILITY

Pars. 7-1.2 through 7-1.5 deal with the hydrodynamic considerations involved in the selection of the details of the hull shape. Of equal or greater importance in the development of the hull shape are the hydrostatic considerations of buoyancy and stability which are covered in the paragraphs which follow.

7-1.6.1 Buoyancy

Under conditions of equilibrium in calm water, the sum of the forces and moments acting on an amphibian are equal to zero. If no external forces are acting, this condition requires that the weight of the displaced fluid (buoyancy) be equal to the weight of the vehicle and that the center of buoyancy (the center of gravity of the displaced fluid) be located on a vertical line which passes through the center of gravity of the vehicle. For a given vehicle weight and center of gravity, the volume of the hull and all appendages must be such that these conditions are satisfied. This is accomplished by appropriately adjusting the hull shape.

7-1.6.1.1 Calculation Procedures

In all studies of buoyancy and stability the basic requirement is the determination, by calculation, of the displaced volume and the center of this volume. The basic calculation procedure is one of integration. Linear dimensions from the lines drawing are integrated analytically, numerically, or mechanically to obtain areas which are integrated by the same procedures to obtain volume. In the event that the hull surface can be defined in terms of simple analytical expressions, the areas may be obtained by substituting the proper numerical limits into the expressions resulting from integrating the original analytical expressions. This is almost never possible because of the complex nature of the hull shape. As a result, it is necessary to use numerical or mechanical integration procedures.

Standard reference books on naval architecture such as Refs. 12, 13, and 14 present several numerical integration procedures. The one most commonly used is due to Simpson and is known as Simpson's rule. First the area under

a curve of y, a function of x, is approximated by

$$\Phi = \int_o^{2h} y dx \approx \frac{h}{3} (y_0 + 4y_1 + y_2) \qquad (7\text{-}1)$$

where

Φ = area under the curve of y between x = 0, and $x = 2h$

h = 1/2 the distance between the limits of integration

$y_{0,1,2}$ = value of the function y at x = 0, h, and $2h$

Note: The units of Φ depend on the units of y and x. If x and y are both given in ft, then Φ is ft^2. If y is in ft^2 and x in ft, then Φ is ft^3.

Simpson's First Rule will give exactly the correct area if y may be expressed as a polynomial of x up to the third order (i.e., $y = a_0 + a_1x + a_2x^2 + a_3x^3$). It may be used to approximate the area under any curve if the interval h is made small enough so that a third degree polynomial can be made to closely approximate y over the interval 0 to $2h$. This is of special importance to buoyancy and stability calculations of an amphibian where discontinuities such as wheel wells may make the approximation of the local hull shape by a third degree polynomial poor. In these cases it is better to use a mechanical integrator or to pass a fair extension of the lines over the discontinuity and obtain the area under the faired curve. The area of the discontinuity can then be determined and subtracted from the area under the fair curve. An example of this procedure is given in Table 7-2.

The first moment of an area about the x = 0 axis is approximated by

$$M_x = \int_o^{2h} yxdx \approx \frac{h^2}{3} \left[(0)y_0 + (1)4y_1 + (2)y_2\right] \qquad (7\text{-}2)$$

The "longitudinal" center of the area is given by

$$\bar{x} = \frac{\int_o^{2h} yxdx}{\int_o^{2h} ydx} = \frac{M_x}{\Phi} \text{ (in units of } x) \qquad (7\text{-}3)$$

The first moment of an area about the y = 0 axis is approximated by

$$M_y = \int_o^{2h} \left(\frac{y^2}{2}\right) dx \approx \frac{h}{6} (y_0^2 + 4y_1^2 + y_2^2) \qquad (7\text{-}4)$$

The "vertical" center of the area is given by

$$\bar{y} = \frac{\int_o^{2h} \left(\frac{y^2}{2}\right) dx}{\int_o^{2h} ydx} = \frac{M_y}{\Phi} \text{ (in units of } y) \qquad (7\text{-}5)$$

These quantities are illustrated in Fig. 7-9.

Where the moment of an area is obtained using Simpson's Rule, caution must be observed with respect to the degree of approximation obtained with a third degree polynomial.

In most problems using Simpson's numerical integrations, more than three ordinates are used (i.e., y_0, y_1, and y_2) to break the curve into smaller intervals in order to achieve greater accuracy in the approximation. The technique to get the total area simply involves adding successive integrations of the type given in Eq. 7-1, thus:

$$\Phi = \int_o^{nh} ydx \approx \frac{h}{3}(y_0 + 4y_1 + 2y_2 + 4y_3 + 2y_4 + \ldots 2y_{n-2} + 4y_{n-1} + y_n) \qquad (7\text{-}6)$$

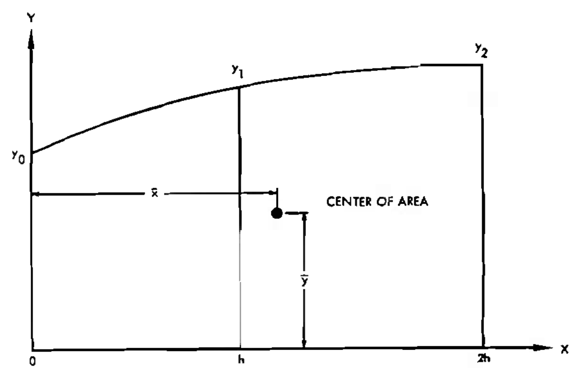


Figure 7-9. Definition Sketch for Area and Center of Area Using Simpson's Integration

where

Φ = area under the curve of y between $x = 0$ and $x = nh$

n = number of equal intervals

h = length of the interval

y_i = value of y at $x = (i)h$

Note: There must be an odd number of ordinates so that n must be an even number.

Similarly,

$$M_x = \int_0^{nh} yx dx \approx \frac{h^2}{3}\Big[(0)y_0 + (1)4y_1 + (2)2y_2 + (3)4y_3 + (4)2y_4 + \ldots (n-2)2y_{n-2} + (n-1)4y_{n-1} + (n)y_n\Big] \quad (7\text{-}7)$$

and

$$M_y = \int_0^{nh} \left(\frac{y^2}{2}\right) dx \approx \frac{h}{6}(y_0^2 + 4y_1^2 + 2y_2^2 + 4y_3^2 + 2y_4^2 + \ldots 2y_{n-2}^2 + 4y_{n-1}^2 + y_n^2) \quad (7\text{-}8)$$

$\bar{x}$ and $\bar{y}$ are still given by Eqs. 7-3 and 7-5, respectively.

The area under a curve on a plane surface may be obtained mechanically using a polar planimeter. This is a simple instrument which measures an area on a plane surface by means of a mechanical polar integration. The theory, operation, and accuracy of polar planimeters are described in Ref. 15. The major advantage of using a planimeter instead of numerical methods for integration is in the possible time saving.

The procedure used to find the buoyancy and center of buoyancy of a hull's underwater volume with Simpson's integrations follows.

First, the sectional area up to the desired waterline is determined at each station, using Eq. 7-6. These areas are then used as the ordinates y_i in Eq. 7-6 with the station spacing = h, to find the displaced volume $\Phi = \nabla$. The buoyancy (force) is found by multiplying the volume by ρg where ρ is the mass density of the displaced fluid and g the gravitational acceleration. Values of ρg commonly used are 64.0 lb/ft^3 for sea water (sw) and 62.2 lb/ft^3 for fresh water (fw), with volume in cubic feet.

The longitudinal center of buoyancy *LCB* is calculated using Eqs. 7-3 and 7-7 in the same manner as for volume.

If it is desired to find the vertical center of buoyancy VCB first, the vertical center of area $\bar{y}$ at each station must be found using Eqs. 7-5 and 7-8. Then Eq. 7-8 is used again except that the terms $y_i^2/2$ are replaced by $y_i\bar{y}_i$ where y_i is the cross-sectional area at station i and $\bar{y}_i$ is the corresponding vertical center of area. The resulting volume moment is divided by the volume to find the vertical center of buoyancy.

An alternative procedure, also widely used, is quite similar. First the waterplane areas for several equally spaced waterline depths are found, and then these are integrated in the vertical direction to find the volume. *LCB* and *VCB* are found using analogous methods.

Finally, corrections are made to the displaced volume and center of buoyancy to account for wheel wells, wheels, and any appendages including shafts, struts, keels, skegs, rudders, propellers, and shrouds.

7-1.6.1.2 Curves of Form

The development of a hull shape that has the necessary buoyancy and center of buoyancy location is a trial and error process. A set of lines is drawn and then the buoyancy, center of buoyancy, and stability are calculated to determine if the equilibrium conditions are satisfied. If the resulting freeboard, draft, trim, or stability are unsatisfactory; the lines are modified and the calculations repeated.

To aid in this process, several sets of curves can be developed. The most useful of these are the Bonjean curves which are a graphical representation of the area of a hull station as a function of draft. As an example, Fig. 7-10 presents a set of Bonjean curves for a LARC V. These curves may be used to determine the buoyancy and longitudinal center of buoyancy location for a waterline at any trim angle.

As an example, in Fig. 7-10 the approximate full load waterline is indicated on the Bonjean curves. The resulting area below this waterline at each station is plotted in Fig. 7-11 as a sectional area curve. The area under this curve is the displaced volume of the hull. This volume, when multiplied by the density of water, is the hull buoyancy. The discontinuities in the sectional area curve are due to the wheel wells. It is important to note that to obtain the total

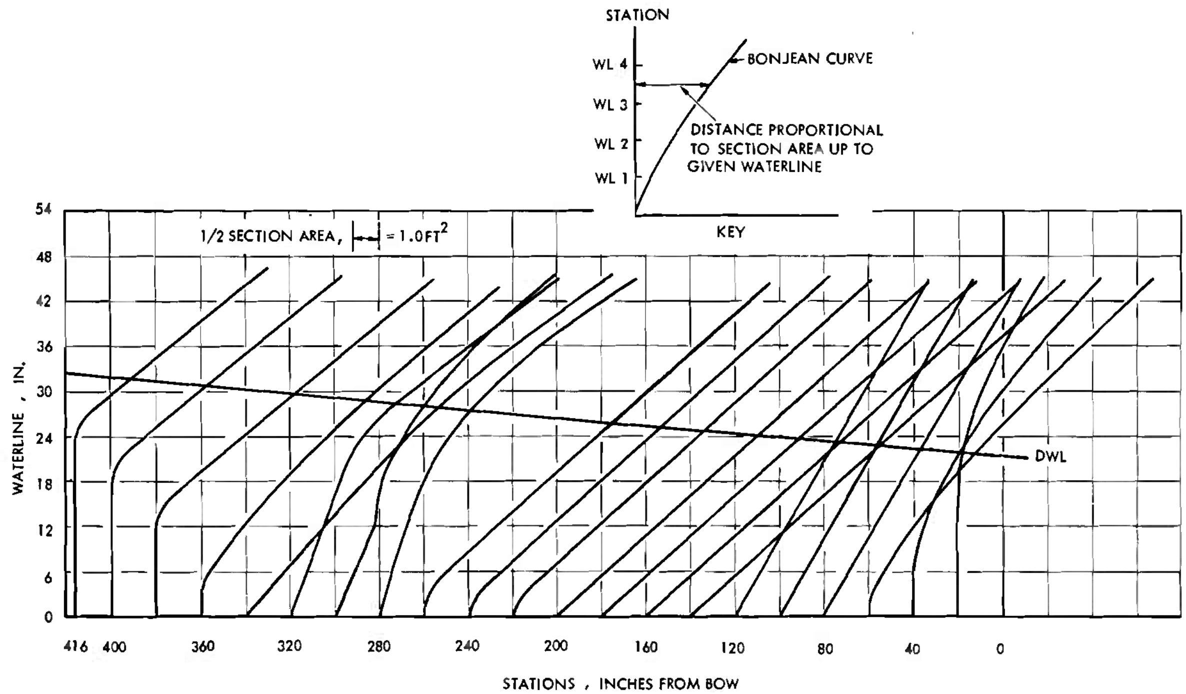


Figure 7-10. Bonjean Curves for LARC V

buoyancy of the amphibian, the displacement of the appendages and wheels must be added to that of the hull. For a LARC V, the wheels account for about 10 percent of the buoyancy.

In the design of a ship or a boat it is common to prepare other curves of form from the final lines. These curves include—as a function of draft—the displacement, wetted surface, vertical and longitudinal center of buoyancy, tons per inch immersion, moment to change trim, and transverse and longitudinal metacentric height. Many of these curves are not used directly in the design of an amphibian, hence, in general, a full set of curves of form is not prepared. Details of the method of calculation of these quantities are given in Refs. 12, 13, and 17.

7-1.6.2 Transverse Stability

A free body at rest and floating in still water is in equilibrium when the sum of all the forces and moments acting on the body are equal to zero. If the body is displaced slightly in any way, this state of equilibrium is (1) stable if the body returns to its original state; (2) neutral if it remains in the displaced position, and (3) unstable if it moves away from the original position. The transverse stability of an amphibian is generally associated with its response to roll, or heel, displacements. In the discussion which follows, stability is taken to mean transverse stability.

An amphibian can have more than one position of stable equilibrium, including one where the vehicle is capsized with the bottom up (if it will remain afloat in this condition). The normal position of stable equilibrium is upright or near upright. In the upright position, the centerline plane of the amphibian is perpendicular to the still water surface and the heel angle θ is defined as zero. It is possible that an amphibian has unstable equilibrium in the upright position due to the condition of cargo loading. In this case, the amphibian would list through a small angle until a position of stable equilibrium is reached.

7-1.6.2.1 Positive Stability

An amphibian has positive stability if the buoyant forces coupled with the weight forces act to return the amphibian to the upright position from a heeled position. The righting moment of a stable amphibian at some heel

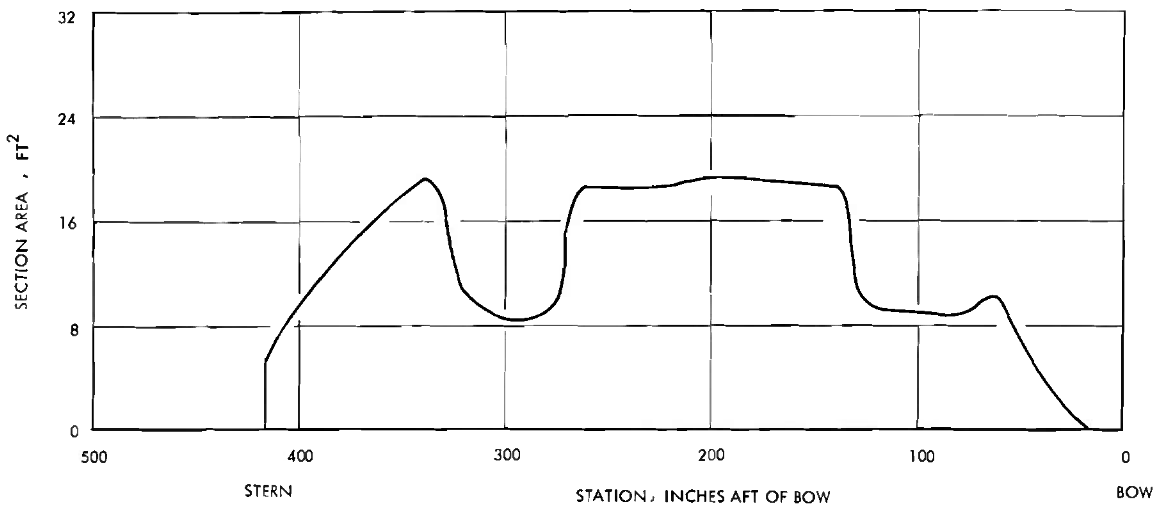


Figure 7-11. LARC V Hull Section Area Curve to Design Waterline

angle is shown in Fig. 7-12 and is equal to the displacement Δ times the righting arm GZ. The righting arm is a function of the heel angle θ and the shape of the hull. Fig. 7-13, a statical stability curve, shows a typical curve of righting arm GZ as a function of heel angle θ. GZ is normally zero at the upright position, increases to a maximum value with increasing heel, and then decreases through zero with further increase in heel.

7-1.6.2.2 Range of Stability

The range of stability is defined as the range of heel angles to one side (port or starboard) over which the righting moment is positive, i.e., acts to return the amphibian to an upright position. This is indicated in Fig. 7-13 as the range over which GZ is positive. Experience indicates that an acceptable range of stability for a fully loaded amphibian with wheels lowered for landing on a beach is from 50 deg to 60 deg. The range of stability is important with respect to the amphibian's ability to operate in surf and heavy seas.

7-1.6.2.3 Dynamic Stability

Dynamic stability at any heel angle is defined as the work done to heel the amphibian to that angle. The dynamic stability can be calculated by integrating to find the area under the statical stability curve (see Fig. 7-13):

$$E_\theta = \frac{\Delta}{57.3} \int_0^\theta GZ d\theta, \text{ ft-lb} \tag{7-9}$$

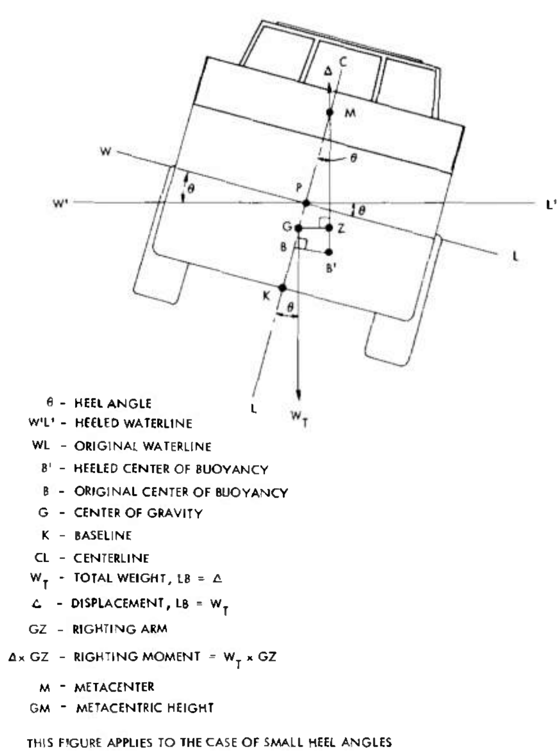


Figure 7-12. Amphibian With Positive Stability

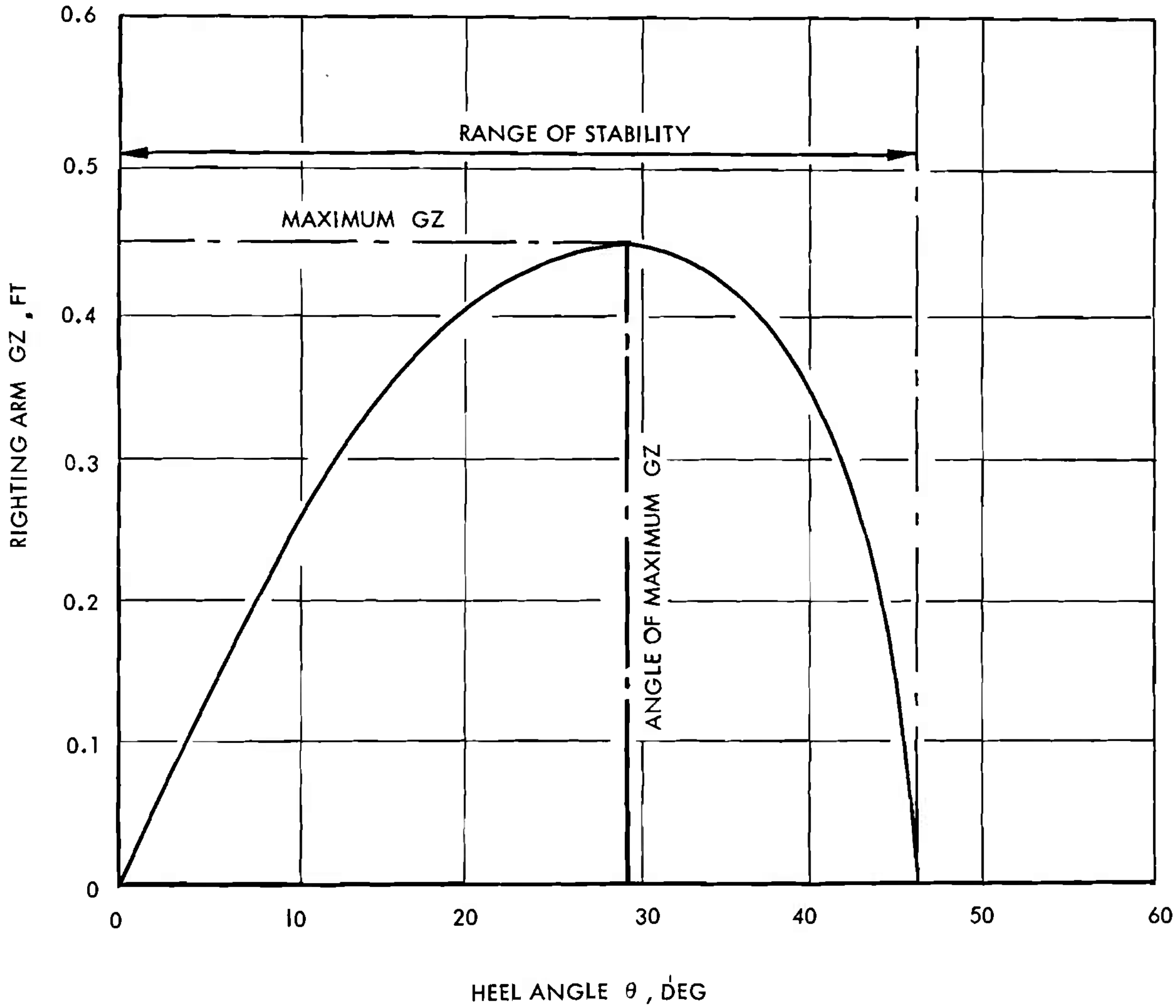


Figure 7-13. Statical Stability Curve

where

E_θ = the work to heel to θ, ft-lb
Δ = displacement, lb
GZ = righting arm, ft (a function of θ)
θ = heel angle, deg

Dynamic stability is rarely calculated for amphibian design, but may be used in investigations of motions in waves. A typical use of dynamic stability would be to calculate the heel that a ship would undergo to absorb the energy from the recoil of a large gun.

7-1.6.2.4 Cross Curves of Stability

The cross curves of stability are determined by calculating the buoyancy and center of buoyancy of the underwater volume at several waterlines for a constant heel angle. The righting arm is given by trigonometry:

$$GZ = CB \sin\theta - (KG - KB') \sin\theta, \text{ ft} \quad (7\text{-}10)$$

where

GZ = righting arm, ft
CB = transverse distance from centerline to center of buoyance B' (BB' in Fig. 7-12 for small heel angles), ft
KB' = perpendicular distance from B' to keel or baseline K (KB in Fig. 7-12 for small heel angles), ft

KG = vertical distance to center of gravity from keel or baseline, ft

θ = heel angle, deg

The righting arm is plotted versus displacement for constant values of heel angles as in Fig. 7-14.

The cross curves of stability are useful for determining the statical stability curve at any desired displacement. Simpson's integration may be used to find the buoyancy and center of buoyancy. However, time may be saved by using simple mechanical methods such as Blom's Method or integrating planimeters. These methods are discussed in Refs. 12, 13, and 14.

7-1.6.2.5 Initial Stability

Initial stability is measured by the metacentric height GM, as in Fig. 7-12, for an infinitesimal heel angle $d\theta$. It may be defined as the slope of the statical stability curve at zero heel angle (Fig. 7-13):

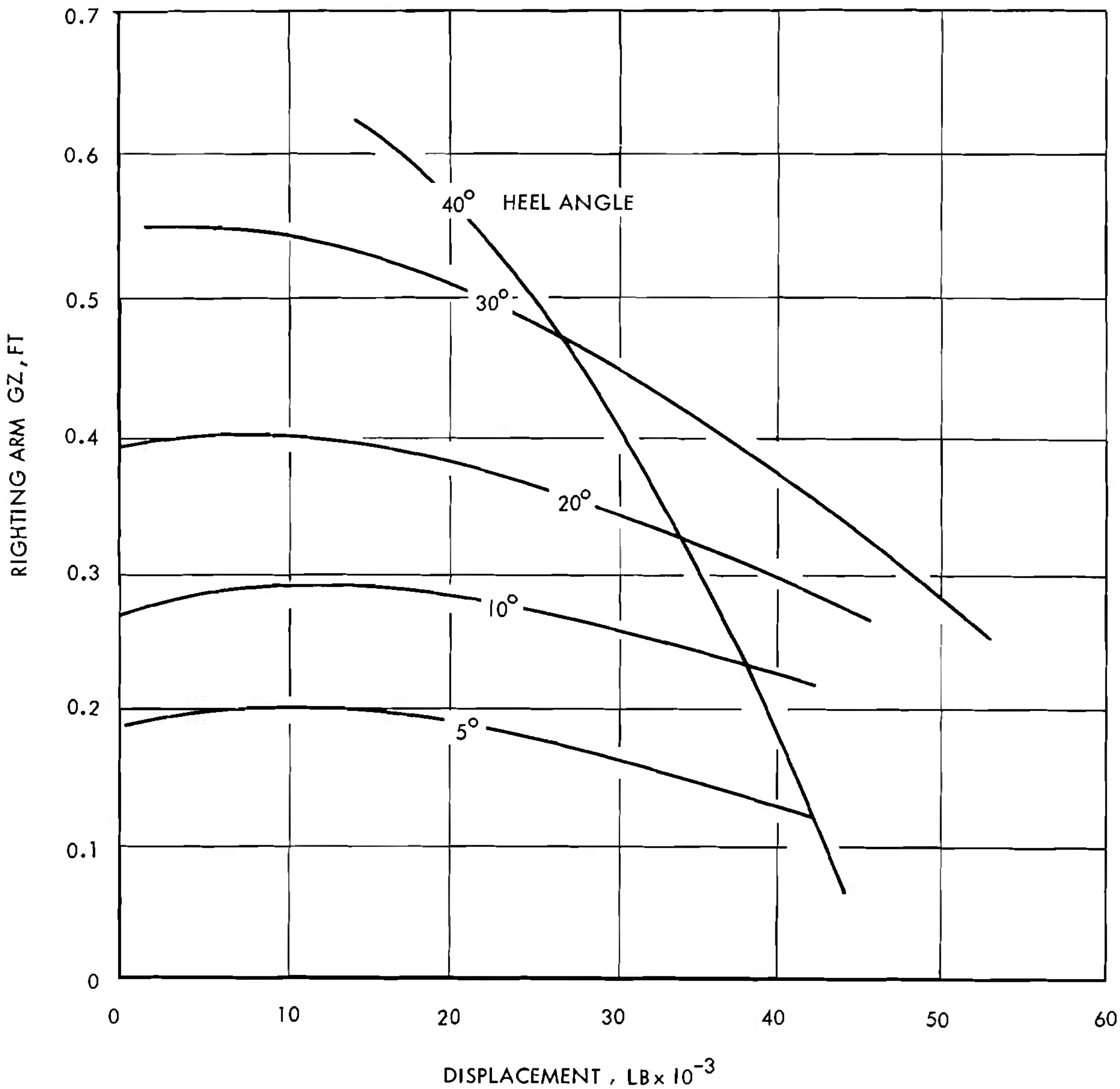


Figure 7-14. Cross Curves of Stability

$$GM = \left.\frac{d(GZ)}{d\theta}\right|_{\theta \to 0}, \text{ ft} \qquad (7\text{-}11)$$

where θ is in radians.

M is the metacenter in Fig. 7-12. The metacentric height is positive when the metacenter lies above the center of gravity. Fig. 7-15 shows the righting arm curve for a vehicle with negative initial stability, i.e., the metacenter at zero heel angle lies below the center of gravity. This particular vehicle would be in stable equilibrium at a heel angle of 11 deg.

A criterion for initial stability, based on experience, is that the metacentric height is at least 0.18 times the beam.

7-1.6.2.6 Calculation of Initial Stability

The initial stability GM is found from simple geometry (Fig. 7-12):

$$GM_t = BM_t + KB - KG, \text{ ft} \qquad (7\text{-}12)$$

where

there are no free surface effects and the subscript t indicates the transverse metacenter

BM_t = vertical height of the transverse metacenter from the center of buoyancy, ft

KB = vertical center of buoyancy from keel or baseline, ft

KG = vertical center of gravity from keel or baseline, ft

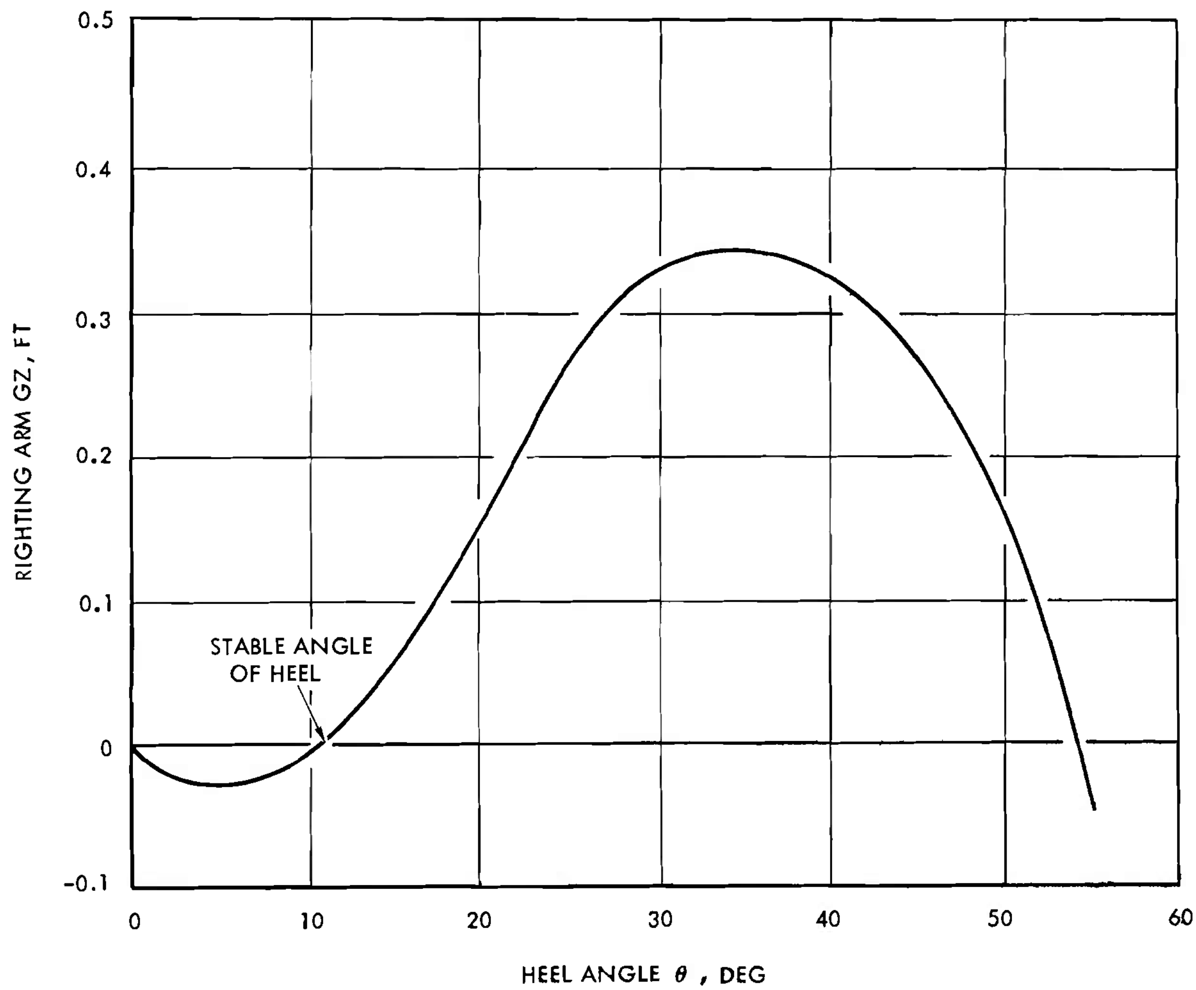


Figure 7-15. Statical Stability Curve, Initially Unstable Vehicle

M_t = point M in Fig. 7-12

The only unknown at this time is BM_t.

Consider the transverse movement of the center of buoyancy from B to B' in Fig. 7-12 due to an incremental change $\Delta\theta$ in heel angle. The total underwater volume remains constant. However, a wedge of volume $L'PL$ extending from the bow to the stern has been immersed and a corresponding wedge WPW' has emerged. The area of wedge at any cross section is given by

$$\tfrac{1}{2}\, y(y \tan \Delta\theta),\ \text{ft}^2 \qquad (7\text{-}13)$$

where

y = the half-breadth of the amphibian at the waterline, ft

The distance between the centers of area of the immersed and emerged wedges is just 4/3 y, ft.

Integrating over the length of the amphibian, the moment of volume about the plane of symmetry is given by

$$2/3\,(\tan \Delta\theta) \int_o^L y^3 dx,\ \text{ft}^4 \qquad (7\text{-}14)$$

Hence, the lateral shift in center of buoyancy CB is:

$$CB = 2/3 \left(\frac{\tan \Delta\theta}{\nabla}\right) \int_o^L y^3 dx = \left(\frac{\tan \Delta\theta}{\nabla}\right) I_t,\ \text{ft} \qquad (7\text{-}15)$$

where

∇ = total underwater volume, ft^3

I_t = (transverse) moment of inertia of the waterline plane about its longitudinal centerline, ft^4

But,

$$BM_t = \frac{CB}{\tan \Delta\theta} = \frac{I_t}{\nabla},\ \text{ft} \qquad (7\text{-}16)$$

Finally,

$$GM_t = \frac{I_t}{\nabla} + KB - KG - \sum\left(\frac{i\rho_t}{\nabla \rho_s}\right) C_{f_s},\ \text{ft} \qquad (7\text{-}17)$$

where

$\sum\left(\frac{i\rho_t}{\nabla \rho_s}\right) C_{f_s}$ = sum of reductions in metacentric height due to free surface effects discussed in par. 7-1.6.3, ft

The importance of waterplane area inertia I_t and vertical center of gravity KG to transverse stability is apparent from this equation.

An example calculation of transverse stability is presented. Volume ∇, vertical center of buoyancy KB, and waterplane inertia I_t are found by using Simpson's integrations. The data presented are for the LARC V.

Full load displacement and vertical center of buoyancy calculation: Eqs. 7-3, 7-6, and 7-7 are used where:

x = waterline height from baseline, ft

y = ½ area of waterplane at x, ft^2

h = 0.25 ft

n = 10

Summation of the data is presented in Table 7-1.

Since ½ areas are used, Eq. 7-6 is doubled to find the total volume:

$$\nabla = \frac{2}{3} h\, \Sigma 1 = \frac{2}{3} (0.25)\ 2581.1 \text{ (read } \Sigma 1 \text{ from Table 7-1)}$$

$$= 430 \text{ ft}^3$$

or

$$\Delta = 430 \times 64.0 = 27{,}600 \text{ lb (sw*)}$$

* salt water

Eq. 7-7 must also be doubled to find the volume moment about the baseline:

$$M_x = \frac{2}{3} h^2\, \Sigma 2 = \frac{2}{3} (0.25)^2\ 13381 \text{ (read } \Sigma 2 \text{ from Table 7-1)}$$

$$= 557 \text{ ft}^4$$

Eq. 7-3 yields

$$KB = \frac{M_x}{\nabla} = \frac{557}{430} = 1.29 \text{ ft}$$

A correction for buoyancy of tires is now made:

TABLE 7-1 SIMPSON'S INTEGRATION FOR FULL LOAD DISPLACEMENT AND KB

1	*2*	*3*	*2 X 3*	*4*	*2 X 3 X 4*
Waterplane, in.	*1/2 Area, ft^2*	*S.M.**	*Product*	*Step*	*Moment*
0 (Baseline)	0	1	0	0	0
3	69.8	4	279.2	1	279.2
6	79.4	2	158.8	2	317.6
9	85.4	4	341.6	3	1024.8
12	92.6	2	185.2	4	740.8
15	97.4	4	389.6	5	1948.0
18	103.7	2	207.4	6	1244.4
21	121.0	4	484.0	7	3188.0
24	106.8	2	213.6	8	1708.8
27	71.9	4	287.6	9	2588.4
30	34.1	1	34.1	10	341.0
			$\Sigma 1 = 2581.1$		$\Sigma 2 = 13381.0$

**S. M. = Simpson multiplier*

Tire size = 18 X 25

Approximate as torus r_{max} = 19 in., r_{min} = 7 in. (from tire manufacturer)

$$\text{Volume per tire} = 2\pi^2 r_{min}^2 r_{max}$$

$$= 2\pi^2 \left(7^2\right) 19$$

$$= 18{,}350 \text{ in.}^3$$

$$= 10.6 \text{ ft}^3$$

$$\left.\begin{array}{l} KB, \text{ fwd} = 4 \text{ in.} \\ KB, \text{ aft} = 2 \text{ in.} \end{array}\right\} \text{Added Moment} = 4\left(\frac{3}{12}\right)10.6 = 10.6 \text{ ft}^4$$

Total volume ∇	=	$(4)10.6 + 430 = 472 \text{ ft}^3$
	Δ =	30,200 lb (sw)
Total M_x	=	$557 + 10.6 = 568 \text{ ft}^4$
Final KB	=	$\frac{568}{472} = 1.20$ ft

The waterplane inertia I_t must be calculated in steps due to the complex shape of the LARC V hull. A Simpson's integration is used for smoothly faired approximation of the waterplane area. Wheel well cutouts and the contribution of the tires are treated as corrections.

A form of integration similar to Eq. 7-8 is used for

$$I_t = \frac{2}{3} \int_0^L y^3 \, dx$$

where y is the half-breadth at the full load waterline, ft. The station spacing h = 20/12 ft. Summation of the data is presented in Table 7-2.

The transverse moment of inertia I_t is given by

$$I_t = \frac{2}{3}\left(\frac{h}{3}\right)\Sigma 3 = \frac{2}{9}\left(\frac{20}{12}\right)5571 \text{ (read } \Sigma 3 \text{ from Table 7-2)}$$

$$= 2060 \text{ ft}^4$$

Wheel well cutouts:

Forward, 32 in. wide X 52 in. long with center 41 in. from centerline

TABLE 7-2 SIMPSON'S INTEGRATION FOR I_t AT FULL LOAD DISPLACEMENT

1 Station	*2* Half-breadth, ft	2^3 $(HB)^3$	*3* S. M.*	3×2^3 Product
0	0.0	0.0	1	0.0
20	0.75	0.42	4	1.68
40	2.25	11.40	2	22.8
60	3.58	45.85	4	183.4
80	4.25	76.75	2	153.5
100	4.50	91.125	4	364.5
120	4.83	112.68	2	225.36
140	4.83	112.68	4	450.72
160	4.83	112.68	2	225.36
180	4.83	112.68	4	450.72
200	4.83	112.68	2	225.36
220	4.83	112.68	4	450.72
240	4.83	112.68	2	225.36
260	4.83	112.68	4	450.72
280	4.83	112.68	2	225.36
300	4.83	112.68	4	450.72
320	4.83	112.68	2	225.36
340	4.83	112.68	4	450.72
360	4.83	112.68	2	225.36
380	4.83	112.68	4	450.72
400	4.83	112.68	1	112.68
				$\Sigma 3 = 5571.12$

S. M. = Simpson multiplier

Aft, 24 in. wide $\times$ 38 in. long with center 46 in. from centerline

Tires:

Assume waterplane geometry to be 10 in. wide $\times$ 20 in. long rectangles 54 in. from centerline

$$
\begin{aligned}
I_{correction} = & - \left[\frac{1}{12}\left(\frac{32}{12}\right)^3\left(\frac{52}{12}\right) + \left(\frac{32}{12}\right)\left(\frac{52}{12}\right)\left(\frac{41}{12}\right)^2\right] \text{(2), front wheel wells} \\
& - \left[\frac{1}{12}\left(\frac{24}{12}\right)^3\left(\frac{38}{12}\right) + \left(\frac{24}{12}\right)\left(\frac{38}{12}\right)\left(\frac{46}{12}\right)^2\right] \text{(2), aft wheel wells} \\
& + \left[\frac{1}{12}\left(\frac{10}{12}\right)^3\left(\frac{20}{12}\right) + \left(\frac{20}{12}\right)\left(\frac{10}{12}\right)\left(\frac{54}{12}\right)^2\right] \text{(4), tires}
\end{aligned}
$$

$$= -284 - 190.6 + 112.8$$

$$= -361.8 \text{ ft}^4$$

$$I_{t_{final}} = I + I_{correction}$$

$$= 2{,}060 - 362$$

$$= 1{,}698 \text{ ft}^4$$

$$\boxed{BM_t = I_t / \nabla = 1{,}698/472 = 3.60 \text{ ft}}$$

The weight and center of gravity of the LARC V are found using the weight summaries of the amphibian. The center of gravity and the weight of the cargo may be estimated.

By taking moments of the two about the baseline, the KG of the loaded amphibian may be calculated:

Item	Weight, lb	Arm, ft	Moment
Light weight	20,200	2.14	43,200
Cargo	10,000	4.62*	46,200
	ΣW=30,200		ΣM=89,400

* Note cargo center assumed 20 in. above deck.

$$\boxed{KG = \frac{\Sigma M}{\Sigma W} = \frac{89{,}400}{30{,}200} = 2.96 \text{ ft}}$$

Finally the initial stability, neglecting free surface corrections, may be calculated:

$$GM_t = BM_t + KB - KG$$

$$= 3.60 + 1.20 - 2.96$$

$$\boxed{GM_t = 1.84 \text{ ft}}$$

7-1.6.2.7 Heeling Forces

There are a number of forces which tend to heel an amphibian including those due to wind, waves, movement of personnel, shifting of cargo, propeller torque, and maneuvering. The discussion which follows will apply only to those forces which may be treated statically to predict equilibrium conditions.

The heeling moment caused by lateral movement of cargo, personnel or other items is given by:

$$\text{Heeling Moment} = Wa\cos\theta, \text{ ft-lb} \qquad (7\text{-}18)$$

where

W = weight of shifted item, lb

a = transverse movement of item's center of gravity, ft

θ = heel angle, deg

In case the cargo is loaded or removed, the heeling moment is given by Eq. 7-18 where a is the transverse distance between the cargo center of gravity and the amphibian's centerline.

The rudder or any other steering device produces turning moments from force components in the transverse direction. At the same time a heeling moment is produced:

$$\text{Heeling Moment} = \Sigma F_t a, \text{ ft-lb} \qquad (7\text{-}19)$$

where

ΣF_t = sum of the transverse components of the hydrodynamic forces, lb

a = vertical arm between the center of forces and the vehicle CG, ft

In a steady turn, the centrifugal force at the vehicle CG is equal to the centripetal force at the center of lateral hydrodynamic pressure. The heeling moment is equal to the centrifugal force times the vertical arm between the CG and the lateral center of pressure of the underwater body:

$$\text{Heeling Moment} = \left(\frac{mV^2}{R}\right)a, \text{ ft-lb} \qquad (7\text{-}20)$$

where

m = vehicle mass = Δ/g, slug

V = linear speed in turn, ft/sec

R = radius of turn, ft

a = vertical arm, ft

The center of pressure may be assumed as the center of the projected area of the profile below the waterline at rest.

A heeling moment is applied to the amphibian by the propeller. This moment is just equal to torque transmitted by the propeller shaft and acts in the direction opposite to the propeller rotation.

The heel angle resulting from any of these moments is determined from statical stability curves. In many cases, the heeling moment is a function of heel angle. For instance, the center of pressure on the underwater portion of the hull in a turn depends on the hydrodynamic flow over the hull, hence, on the radius of turn and on heel angle. The heeling moment may be resolved into a heeling arm by dividing by the amphibian's displacement. This heeling arm may then be plotted with the statical stability curve. The point at which the two curves cross is the equilibrium position, i.e., the heeling arm is equal to the righting arm as in Fig. 7-16.

7-1.6.2.8 Stability Criteria

No suitable theory for predicting the forces acting on an amphibian in surf is available at this time. In the past, criteria for adequate stability of amphibians have been limited to simple requirements on range of stability and initial stability based on experience. Some additional stability criteria, taken from Ref. 16, are presented here. The criteria were established as guidelines for U.S. Navy ships and may prove useful for future amphibian designs. These criteria may be used to assure that adequate reserve stability is provided when the amphibian is heeled over due to any of the types of forces presented in par. 7-1.6.2. See also Ref. 17.

The worst probable heeling condition should be calculated whether this be caused by poor location of the cargo, high-speed turns, or any other likely situation. The suggested criteria for adequate stability in this condition are (Fig. 7-16):

a. The steady heel angle does not exceed 15 deg

b. The righting arm at the steady heel angle does not exceed 60% of the maximum value of righting arm.

c. The reserve dynamic stability represented by the shaded area in the figure is not less than 40% of the total area under the righting arm curve.

The reserve stability indicated by these criteria is required to provide a margin against capsizing due to the possible concurrence of more than one of the conditions that have been discussed, including the effects of wind and wave actions. It should be noted that the amphibian is subject to capsize if the heeling arm should exceed the maximum righting arm or if the heel angle should exceed the range of stability.

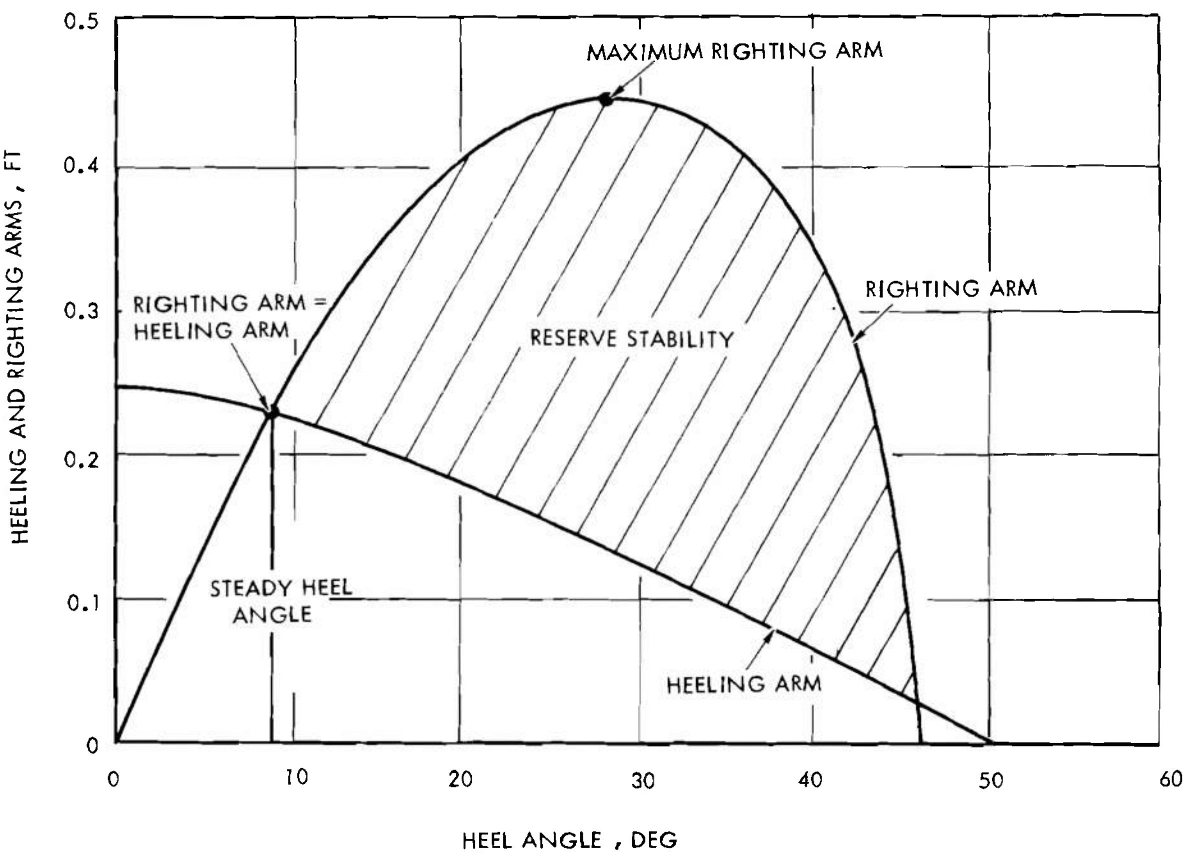


Figure 7-16. Stability Criteria

7-1.6.2.9 Grounded Stability

Coming ashore through surf and waves, an amphibian may drive onto a shoal, reef, or ledge, thereby grounding. Grounding results in a virtual rise in the center of gravity and may seriously degrade its transverse stability. The virtual or effective center of gravity is given by:

$$KG_v = \frac{KG(\Delta) + QK(F)}{\Delta - F}, \text{ft} \quad (7\text{-}21)$$

where

KG = height of center of gravity above baseline, ft
Δ = amphibian displacement, lb
F = grounding force, lb
Q = intersection of the grounding force vector with the amphibian centerline plane
QK = height along centerline from Q to the baseline, ft

Subscript v denotes virtual or effective value.

The virtual displacement of the grounded amphibian is

$$\Delta_v = \Delta - F, \text{lb} \quad (7\text{-}22)$$

For a free floating amphibian at small heel angle (Fig. 7-11)

$$\text{Righting Arm } GZ = GM \sin\theta, \text{ft} \quad (7\text{-}23)$$

and

$$\text{Righting Moment} = GZ\,(\Delta), \text{ft-lb} \quad (7\text{-}24)$$

Similarly, for a grounded amphibian (Fig. 7-16)

$$\text{Righting Arm } G_vZ_v = G_vM' \sin\theta, \text{ft} \quad (7\text{-}25)$$

and

$$\text{Righting Moment} = G_vZ_v(\Delta_v), \text{ft-lb} \quad (7\text{-}26)$$

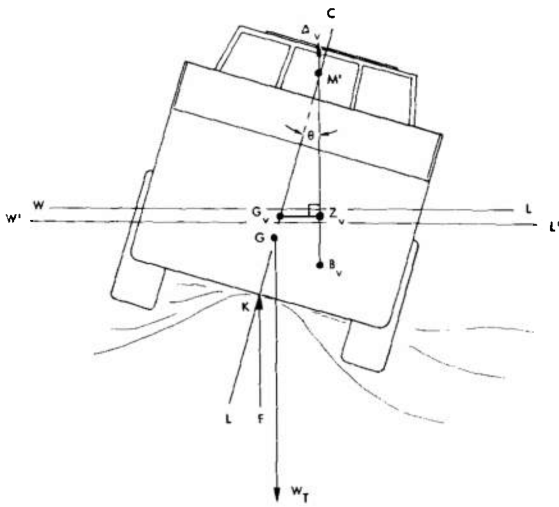
where M' is the metacenter of the grounded waterline as shown in Fig. 7-17.

In Fig. 7-17 the point Q is at the baseline so that $QK = 0$, as shown. In a more general case, Q will lie below the baseline and is a function of the heel angle θ. The amphibian is stable at some angle θ and the waterline $W'L'$ if the virtual center of gravity lies below the metacenter M'; i.e., a positive righting moment exists.

A criterion which is suggested here for adequate stability in the grounded condition is that a positive righting moment be maintained for any waterline when the amphibian is grounded on a side slope slightly less than the requirement for side slope stability on land.

7-1.6.2.10 Damaged Stability

The stability of an amphibian which has incurred hull damage below the waterline is degraded due largely to the free surface effects. The subsequent flooding results in a virtual decrease of waterplane area and lowers the metacentric height. Recent amphibians do not have watertight compartmentation because of the added weight that would result from this safety feature. As a result, the entire hull is subject to flooding from damage at any point unless the bilge pumps are capable of pumping water at a rate equal to or higher than it enters



WL - HEELED, UNGROUNDED WATERLINE
W'L' - HEELED, GROUNDED WATERLINE
B_v - CENTER OF BUOYANCY, GROUNDED
G - CENTER OF GRAVITY
G_v - VIRTUAL CENTER OF GRAVITY
K - BASELINE
W_T - WEIGHT OF CRAFT = Δ
F - GROUNDING FORCE
Δ_v - VIRTUAL DISPLACEMENT TO W'L'
θ - HEEL ANGLE
M' - METACENTER FOR W'L' AND θ

Figure 7-17. Grounded Amphibian Stability

the hull. Thus, it is only a matter of time before the damaged craft will sink. The question of damaged stability is, in this case, only of interest from a standpoint of how long the craft will remain upright for the crew to abandon ship. In any event, means of escape from a capsized amphibian must also be provided because of the possibility that an amphibian may turn over abruptly in heavy seas or surf. Treatment of damaged stability is fully discussed in Ref. 14.

7-1.6.3 Free Surface Effects

A liquid with a free, unrestrained surface such as fuel in the fuel tanks or water in the bilges of the amphibian will shift when the craft heels so that the center of gravity of the liquid moves, thus creating a heeling moment which opposes the righting moment due to waterplane inertia I_t. In effect, the waterplane inertia is decreased by the inertia of the free surface area in question corrected for the specific gravity of the liquid. Thus, the reduction in metacentric height BM due to a liquid free surface is given by

$$\text{Reduction in } BM = \frac{i\rho_t}{\nabla \rho_s} C_{f_s}, \text{ ft} \qquad (7\text{-}27)$$

where

i = moment of inertia of the free surface area about a longitudinal axis through the center of area of that surface, ft^4

∇ = displaced volume of the amphibian, ft^3

ρ_t = mass density of the liquid, slug/ft^3

ρ_s = mass density of water displaced by the hull, slug/ft^3

C_{f_s} = free surface coefficient, nd

The free surface coefficient C_{f_s} is applied to the moment of inertia when the liquid level is either very low or very high in a tank to take account of the reduction in breadth of the free surface when the tank is heeled slightly. Fig. 7-18 gives values of C_{f_s} for rectangular tanks. Generally, the worst case is calculated for the free surface reduction in metacentric height. In this case, the tank is half-full of liquid and C_{f_s} = 1.0.

For example, consider two fuel tanks, half-full of Diesel oil in the LARC V amphibian of the problem in par. 7-1.6.2.6.

Tank dimensions: (2) length ℓ = 2.5 ft
breadth b = 2.25 ft

Diesel oil: ρ_t = 1.675 slug/ft^3
salt water: ρ_s = 1.99 slug/ft^3
coefficient: C_{f_s} = 1.0
volume: ∇ = 472 ft^3

free surface area inertia, (each tank)

$$i = \ell b^{3/12}$$
$$= 2.5(2.25)^{3/12} = 2.59 \text{ ft}^4$$

free surface reduction in BM_t for both tanks $= 2\left(\frac{2.59}{472}\right)\left(\frac{1.675}{1.99}\right) = 0.01$ ft

The reduction in metacentric height from these fuel tanks is rather small. Larger effects may arise from water in the bilges, cargo well, or on a cargo deck. In these cases, the displaced volume and the amphibian's center of gravity must be corrected for the weight and center of the water acting as if it were added, solid cargo. The free surface effect is calculated as for the fuel tanks.

Free surface effects in tanks can be reduced by fitting liquid-tight longitudinal structures to reduce the effective breadth of the free surface area and, thus, the inertia i.

7-1.6.4 Longitudinal Stability and Trim

Longitudinal stability is not usually a problem. Longitudinal metacentric height GM_ℓ may be calculated in the same manner as transverse GM_t; the only differences being in the moment of inertia of the water plane and in the free surface corrections. Both I_ℓ and i_ℓ are taken about transverse axes.

The waterplane moment of inertia should be taken about the transverse axis through the longitudinal center of area of the waterplane. The axes of individual free surfaces, for corrections, should each be taken at its center of area.

$$GM_\ell = KB + BM_\ell - KG - \sum\left[\frac{i_\ell}{\nabla}\left(\frac{\rho_t}{\rho_s}\right)\right] C_{f_{s\ell}}, \text{ ft} \qquad (7\text{-}28)$$

It is evident that KB and KG are the same in the longitudinal direction as in the transverse direction.

Continuing our example for the LARC V, the longitudinal moment of inertia I_ℓ can be approximated by taking the area inertia about

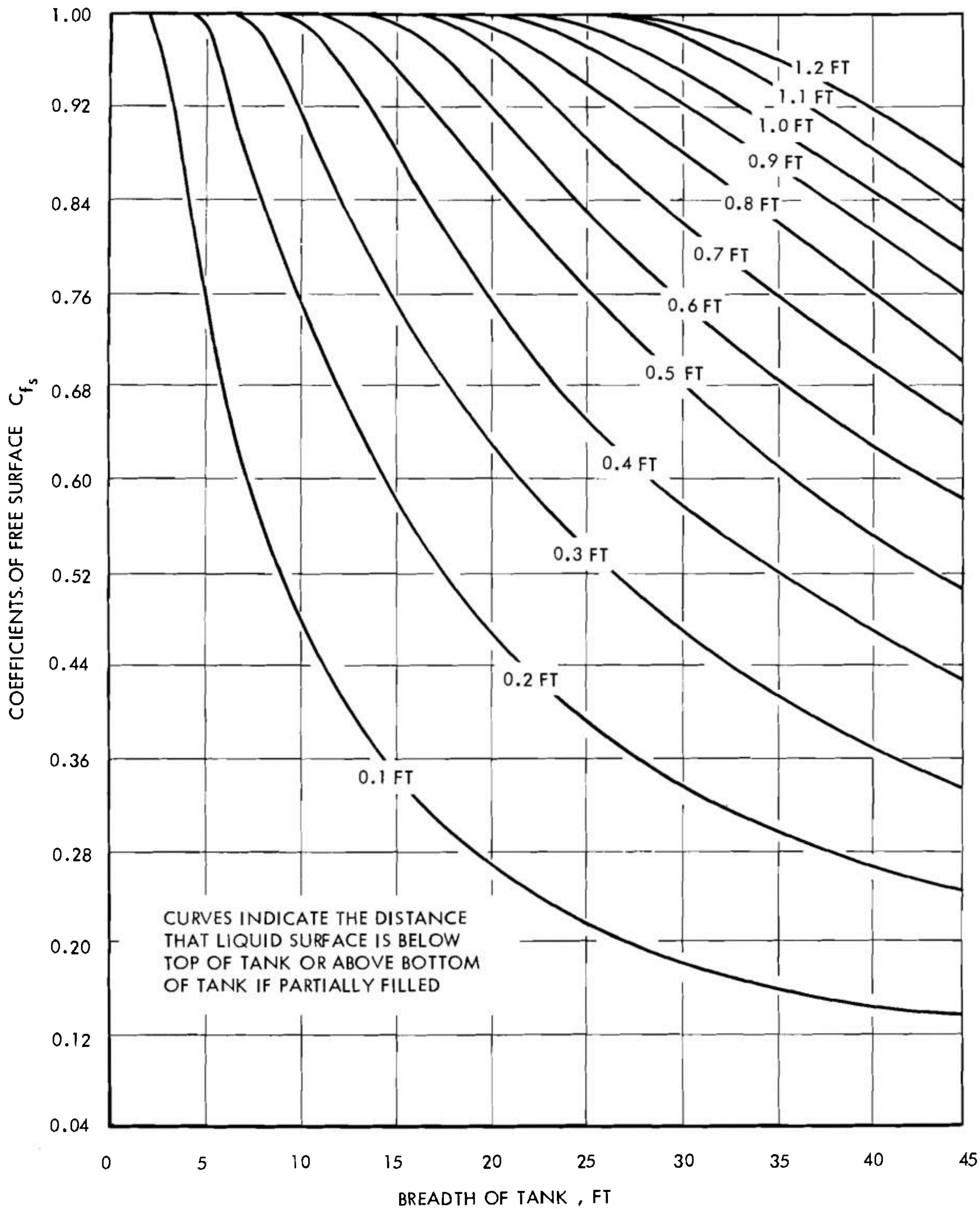


Figure 7-18. Coefficients of Free Surface for Rectangular Tanks When Inclined to 5° Heel

the midpoint of a rectangle which approximates the hull shape.

$L = 31$ ft
$B = 10$ ft

$$I_\ell = \frac{L^3B}{12} = \frac{(31)^3 10}{12}$$

$$= 24{,}800 \text{ ft}^4$$

$\nabla = 472 \text{ ft}^3$

$$BM_\ell = \frac{24{,}800}{472}$$

$$= 52.5 \text{ ft}$$

$$\left.\begin{aligned} KB &= 1.20 \text{ ft} \\ KG &= 2.96 \text{ ft} \end{aligned}\right\} \text{ from par. 7-1.6.2.6}$$

The free surface of the fuel tanks will have very little effect and can be assumed neglected.

Longitudinal Metacentric Height by Eq. 7-28 is

$$GM_\ell = 1.20 + 52.5 - 2.96$$

$$GM_\ell = 50.7 \text{ ft}$$

Thus, the longitudinal metacentric height is considerably greater than the transverse metacentric height.

Trim is defined as the difference between the drafts measured forward and aft. If the draft aft is the greater, the craft is trimmed by the stern; if the draft forward is the greater, the craft is trimmed by the bow.

For an amphibian, it is desirable that the rear wheels touch bottom first while approaching the shore. With the rear wheels on land, the craft will be less likely to be swung around by incoming waves. Trim by the stern should be designed into an amphibian, with due regard given to keeping the cargo deck above water.

The moment required to alter the trim by 1 in. is:

$$MT1 = \frac{\Delta GM_\ell}{12L}, \text{ ft-lb} \qquad (7\text{-}29)$$

where

Δ = displacement, lb
GM_ℓ = longitudinal metacentric height, ft
L = length between points where trim is measured, ft

The subscript ℓ indicates longitudinal direction.

Therefore, the moment required to change the trim of the LARC V by 1 in. is (where L = 23.3 ft)

$$MT1 = \frac{30{,}200(50.7)}{12(23.3)}$$

$$= 5{,}500 \text{ ft-lb}$$

A 175-lb man moving from the bow to the stern will change the trim less than 1 in.

At large angles of heel, trim may result from a change in longitudinal center of buoyancy. This problem can be avoided by designing the vehicle with sufficient freeboard at both bow and stern to provide surplus buoyancy at each end at large heel angles. Thus, the center of buoyancy when heeled does not tend to move longitudinally.

7-1.6.5 Freeboard

Freeboard is the distance measured vertically from the waterline to the deckline. Generally speaking, increased freeboard results in greater reserve buoyancy, greater range of stability, and less likelihood of waves breaking over the deck.

Ease of cargo handling may be of prime consideration to the amphibian designer. The highest reach of a fork-lift truck or the lack of good unloading facilities may dictate a low cargo deck. As a result, the freeboard amidships may be low. Raising the freeboard forward and aft will provide the necessary reserve buoyancy and range of stability without increasing the difficulty of unloading cargo. As long as the hull is watertight and any water on the cargo deck will drain freely, low freeboard amidships will cause no particular problem.

Removable sidewalls in way of the cargo deck may be used to raise the freeboard. As they are not particularly strong, their effectiveness in increasing reserve buoyancy and range of stability is not very great. However, the protection afforded the cargo may make such sidewalls desirable.

7-1.7 LAND OPERATION CONSIDERATIONS

The requirements for operation on land have some effects on the design of an amphibian's hull shape. These effects are basically limitations on principal dimensions, ground clearance, and the approach, break, and departure angles. These limitations are almost always identified during

the preliminary design phase and, as a result, are covered in par. 6-4.2. The size and location of the wheels and wheel wells also affect the hull shape. The size and location of the wheels are determined by the limitations on dimensions, critical angles, and mobility considerations covered in par. 7-8. The size and shape of the wheel wells are determined by the tire size and clearances for steering. The equations relating turning radius to vehicle geometry and wheel angles are presented in par. 7-9.4.

7-2 GENERAL ARRANGEMENT

The detailed arrangement plans serve as a final check prior to manufacturing that the components will fit as planned and that there is sufficient clearance for installation and maintenance. The arrangement drawings should provide sufficient detail and instructions so that all components and systems will be properly installed.

7-2.1 CARGO

The primary mission of a logistic amphibian is to carry cargo. Each phase of the cargo-carrying operation must be considered for each anticipated mix of cargo. This includes:

a. Methods of loading and unloading each item

b. Interference of loading method from cargo previously loaded

c. Securing cargo in all types of weather

d. Trim and stability at all times during loading

e. Protection of crew and parts of the vehicle from swinging cargo

f. Jettisoning cargo

g. Fire in cargo

h. Crew access to machinery with cargo aboard

i. Efficiency of loading cargo, including practical clearances

j. Water drainage with cargo aboard

k. Protection of cargo from the environment

l. Crew escape routes if cargo shifts

m. Tie down considerations (no projections which can be damaged or cause damage)

7-2.2 MACHINERY

In the preliminary design phase the machinery and the associated components are arranged but the accessories, in general, are not considered in any detail. The detailed machinery arrangement should include all components, their location, and method of attachment. The following detail considerations are of importance:

a. Kits for special operating conditions
b. Maintenance procedures
c. Access
d. Air supply system
e. Exhaust system
f. Expansion
g. Spare parts
h. Fasteners
i. Vibration
j. Noise
k. Starting
l. Control
m. Fail-safe features
n. Cooling/heating
o. Clearances
p. Securing procedures
q. Ventilation

The quality of machinery maintenance is controlled to a large extent by the ease with which the work can be accomplished. A high degree of maintainability requires good access and sufficient clearance for each preventive maintenance action and repair which must be accomplished.

Gas turbines require special design consideration for both the intake and exhaust. Both require extra space because of the large air volume required, the removal of salt particles in the intake, and the necessary noise controls.

7-2.3 AUXILIARY SYSTEMS

Many of the auxiliary systems are neglected in the earlier phases and must be studied during final arrangement planning. Some of these systems will be fitted after the major components are installed and must be located accordingly. The design of each system is described in par. 7-10 and there are arrangement requirements associated with each system that are part of the design. Special arrangement precautions that are associated with many of these systems are:

a. Maintainability
b. Operation in environmental extremes
c. Fire fighting
d. Flooding
e. Emergency operation

7-2.4 STRUCTURE

The direct and indirect framing causes many problems in the arrangement. The depth of this structure can be reduced only at the expense of added structural weight. For shapes such as angles, tees, or channels, holes may be cut in the webs with only a minimal loss in strength. The U. S. Navy practice for access and lightening holes in framing of steel ships abstracted from Ref. 18 is:

> Openings in strength members shall be well separated and not aligned in a transverse plane. Where openings, of necessity, are close together, they shall be combined to form a single opening. In general, holes in structure shall be circular. Where a circular opening is not practical, corners of openings in areas of stress (including shear stress) shall be rounded to radii of at least one-eighth (preferably one-fourth) of the clear dimension normal to the direction of the principal stress. If the size or location of an opening impairs the strength of an important structural member, it shall be reinforced.
>
> In general, framing which is six inches or more in depth, shall be lightened except where this would impair the strength such as framing over a stanchion or where the principal stress is parallel to the frame or where the frame is part of a watertight compartment.
>
> Except for special situations, lightening or access holes should be centered at the neutral axis of the combined cross section of the structural members and associated plating. The depth of the hole should not exceed 40 percent of the depth of the web and the length of the hole should be parallel to the member and not more than twice as long as it is deep. Holes shall be spaced so that the distances between edges of adjacent holes will not be less than 5/4 times the length of the holes.
>
> Drain holes and water courses should be provided to permit the free flow of liquids in the bilge and in other pockets to the bilge pumps. In compartments fitted with bilge pumps, the total area of drain holes through any frame or longitudinal shall be at least twice the area of the suction pipe.

Air holes in the framing should be provided to prevent the formation of gas pockets. For amphibians this is generally not a problem except in the case of the battery where hydrogen is generated as the battery is charged.

Each construction material has certain deficiencies which must be specially handled. The arrangement designer should determine from the structural designer what these are for each design. In welded aluminum, for example, the heat affected zone (HAZ) in way of the weld may have a much lower strength than the parent metal so that brackets and clips should not be indiscriminately specified in high stress areas. Likewise, holes for mechanical fasteners must be in areas of low stress.

7-2.5 BUOYANCY, TRIM, AND STABILITY

In order to achieve satisfactory performance in a seaway and survival in bad weather, it is necessary to have a proper relationship between the amount and distribution of weight and of buoyancy. In particular this includes:

a. Excess buoyancy of ends
b. Trim in all conditions of loading
c. Freeboard
d. List in all conditions of loading
e. Change of trim as fuel is burned
f. Initial stability for all conditions of loading and wheel positions (if retractable)
g. Range of stability for all conditions of loading and wheel positions (if retractable)

One problem in this regard peculiar to wheeled amphibians is the wheel wells when they are fitted with side covers. If the wells are not vented, they may catch air and be buoyant or fill with water and add weight when rolling. The trim, buoyancy, and stability calculations must be considered in the worst condition from the standpoint of safety. There are two ways to prevent this—(1) pump air into each wheel well so it is always air-filled, or (2) design wheel wells so they are inherently self-venting. Small vent pipes are not sufficient because they will fill with dirt and/or ice:

7-2.6 SAFETY

The safety of each system is the responsibility of the system designer. The design of the arrangement must consider the interrelationship of each system with the others and ensure that the total design will be safe to operate.

7-2.7 HUMAN FACTORS ENGINEERING

The human factors engineering for an

amphibian must take into consideration that these vehicles are used to the limit of the ability of the man-machine system to function. They will be used in a war-time environment for missions for which they are not designed, by people who may have not been properly trained, and in an environment exceeding the original specification. From this point of view it is obviously necessary to design the vehicles so that standard operations can be accomplished with only average human ability.

Ref. 19 is a Human Engineering Laboratories Standard for use by the subordinate commands of the Army Materiel Command (AMC) in the area of human factors engineering, in accordance with Suppl. 1 to AR 602-1. It gives engineering design principles and detailed criteria for anthropometrics and the environment. Refs. 20-22 provide additional data on human response to vehicle vibration.

7-3 HULL STRUCTURE

Amphibians are subjected to a complex assortment of loads including both those associated with wheeled vehicles and with boats. Consequently the structural design cannot be based on past practice for either type of vehicle but must consider all of the forces which will be imposed in conjunction with the limited amount of previous experience available.

This paragraph includes information on:

a. Imposed forces
b. Materials
c. Structural configuration
d. Detail design
e. Hull weights

The primary emphasis is on delineating the problems and methods used for the structural design.

7-3.1 FORCES

The first step in the hull structural design is to estimate, with a satisfactory degree of accuracy, all of the forces which the hull must resist. There are instances where either the forces are so small that they can be neglected or they are so large that no attempt is made to produce a structure which will not be damaged. For example, the former might be a man walking on the cargo deck, and the latter might be the deck being hit by an armor-piercing round. These types of forces can be neglected but the effects of all others that may be critical must be considered.

The forces are divided into the following types:

a. Longitudinal. Vertical forces acting on the hull such that the hull acts as a beam.

b. Torsional. Forces acting perpendicular to the centerline of the hull which tend to twist it around a longitudinal axis.

c. Local. Forces acting on a section of the hull which do not produce a significant stress throughout a major portion of the hull.

For each of these types of forces the load can be applied at different rates which cause the structures to react differently. The different strain rates have been categorized as

a. Creep	10^{-8}–10^{-5}; (in.)/(in.-sec)
b. Static	20^{-5}–10^{-1}
c. Rapid loading	10^{-1}–10^{1}
d. Impact	10^{1}–10^{4}
e. Hypervelocity impact	10^{4}–10^{7}

Some materials will elongate if subjected for a long period of time to a stress below their elastic limit, particularly at high temperatures. This is known as creep and is not important for amphibians. Some aluminum alloys creep at moderate temperatures but not the alloys that are to be used in the marine industry.

Static loads are easiest to handle as no dynamic motion is involved. Many "quick" engineering solutions involve approximations which reduce the answer to the static load condition.

Rapid loading requires only relatively simple factors to account for the dynamics of the loading, but fatigue must be carefully handled for a properly engineered structure.

At the speed of impact loadings (10^{-1} to 10^{-4} sec) materials react differently. The stress-strain curves, in particular, are altered and this change must be considered. The stress-strain relationship above the elastic limit may be increased as much as 50 percent for steel, and 0 to 1 percent for aluminum. Data for impact loadings are not yet available for all materials. Ref. 23 is a basic reference on this subject.

Hypervelocity impact loading can alter the material response severely and in ways that are

not yet predictable. The amphibian structure is not involved with these phenomena.

7-3.1.1 Longitudinal Forces

7-3.1.1.1 Waterborne

The scantlings of longitudinal structural members for ship hulls are normally sized to resist the bending moment imposed on the hull in waves either in a hogging condition when the wave crest is amidships or for the sagging condition when the wave crests are at the ends of the ship. For ships less than about 100 ft in length, the local loads usually are the determining factors for the selection of longitudinal scantlings and not the bending moment. The bending moment stress must be considered, however, as it is at times superimposed on other stresses in the hull and it can be very significant for hulls that are shallow amidships.

The determination of the bending moment is particularly critical for large ships and there has been a substantial amount of research effort expended to improve the accuracy of the prediction techniques. For use on amphibian design, however, the treatment in Ref. 14 is sufficient.

To determine the static bending moment, the hull is considered at the various loading and trim conditions in still water. For the dynamic bending moment it is considered in a stationary wave with a length equal to the waterline length of the hull. For a "sagging" condition where the deck is in compression the wave trough is amidships, and for the "hogging" condition where the deck is in tension the wave crest is amidships. A trochoidal wave profile is normally used with the wave height selected from past experience. For amphibians a wave height of 1.1 times the square root of the waterline length can be used as a satisfactory approximation.

The bending moment calculation consists of:

a. Balancing the hull in the water, either smooth water or in the stationary wave. Bonjean curves are used to determine the buoyancy at selected stations along the hull (see par. 7-1).

b. Calculating the buoyancy along the length of the hull

c. Estimating the weight along the length of the hull

d. Calculating the bending moment along the length of the hull

If desired, the shear and the deflection from this loading can also be determined. Since a high degree of accuracy is not required, the calculation can be simplified by dividing the hull into a number of equal lengths and assuming the buoyancy and weight are constant for the length of each section.

Figs. 7-19 and 7-20 show the curves developed from such a calculation for the LARC V. The maximum bending moment is 34,350 lb-ft, 203 in. aft of the bow for the sagging condition using a wave 2.1 ft high. To estimate the bending moment for similar vessels of other sizes it can be assumed that (Ref. 14):

$$M_B \propto B(LWL)^2\, H_m\, C_B \tag{7-30}$$

where

M_B = bending moment, ft-lb
B = maximum molded breadth at the waterline, ft
LWL = length at the waterline, ft
H_m = mean draft, ft
C_B = block coefficient, nd

7-3.1.1.2 Landborne

Another longitudinal force that must be considered is the land mode. The static bending moment can be calculated in the same manner as for the afloat condition except that the forces supporting the vehicle are simply the loads from the wheels. The dynamic bending moment applied to the hull can be approximated by assuming:

a. A maximum acceleration of the center of gravity,

b. that this acceleration is the result of vertical motion of the wheels farthest away from the center of gravity, and

c. that a vehicle pivots around the remaining set of wheels.

The assumptions are shown in Fig. 7-21. Let

$\triangle W$ = weight of the vehicle, lb
F_F = applied vertical force at the forward wheels (up is positive), lb
F_A = applied vertical force at the aft wheels (up is positive), lb
a = distance from aft wheels to the longitudinal center of gravity, ft
b = distance from the forward wheels to the longitudinal center of gravity, ft

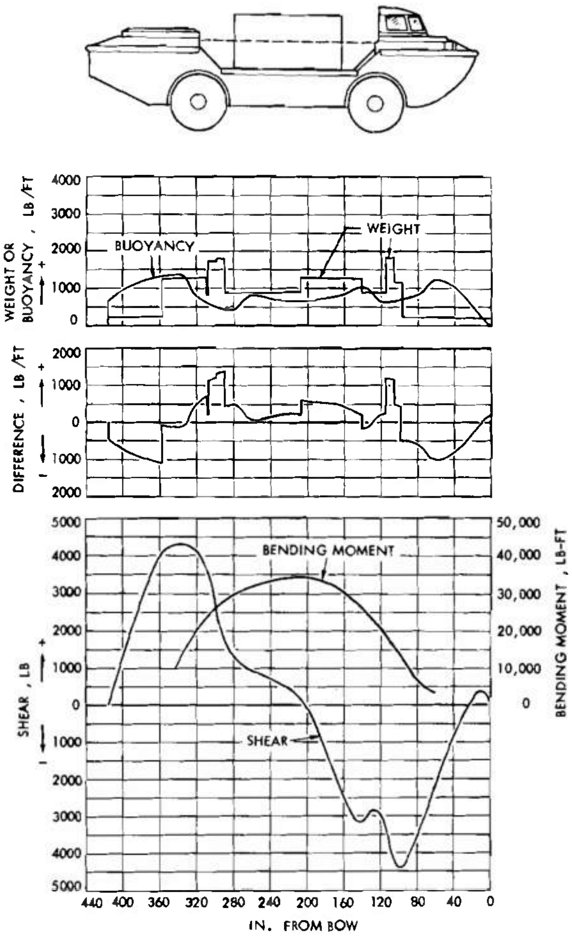


Figure 7-19. Bending Moment Curves for LARC V (Sagging)

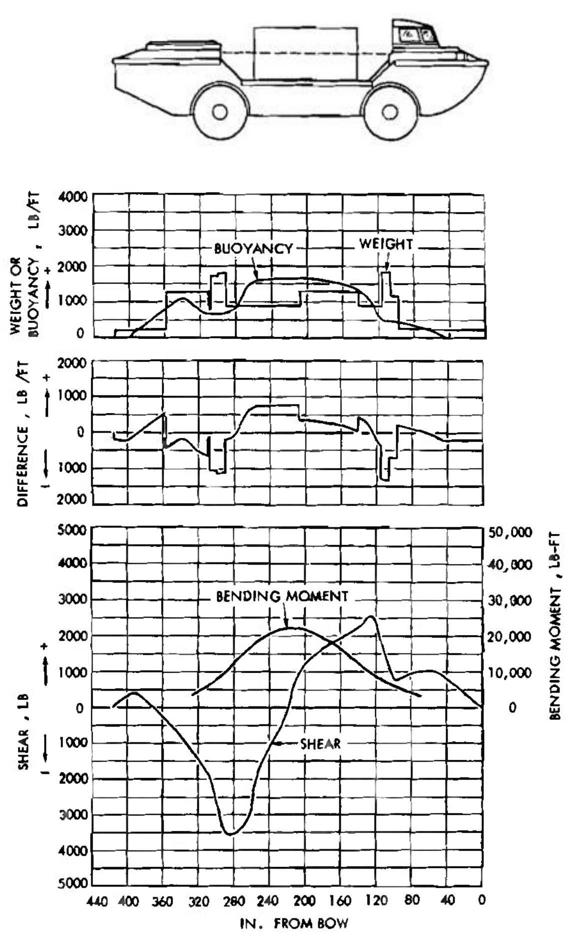


Figure 7-20. Bending Moment Curves for LARC V (Hogging)

ℓ = length of wheelbase, ft

w_x = weight of the vehicle in the x section, lb

x_n = distance to center of x section measured from bow, ft

$\ddot{y}$ = vertical acceleration at the center of gravity, in number of g's, i.e., ft/sec^2 divided by 32.2 ft/sec^2

k = radius of gyration about transverse horizontal axis through the center of gravity, ft

To calculate the bending moment:

1. Determine the weight curve using a number of sections. The weight is to be considered evenly distributed in each section. Fig. 7-22 shows a curve for the LARC V.

2. Plot the acceleration curve. The vehicle is assumed to pivot about the set of wheels nearest to the center of gravity, and the design maximum vertical acceleration at the center of gravity is used. The acceleration curve is a straight line since the amphibian is assumed to be a rigid body. Fig. 7-22 shows a plot for the LARC V.

3. Find F_F from

$$F_F = \frac{\Delta a}{\ell}\left\{\ddot{y}\left[\left(\frac{k}{a}\right)^2 + 1\right] + 1\right\}, \text{lb} \quad (7\text{-}31)$$

4. Determine the bending moment curve for the desired section of the vehicle. This is generally taken at the 'midship section for half of the length. It should include the length of any midship well. The bending moment at section x is calculated using Eq. 7-32:

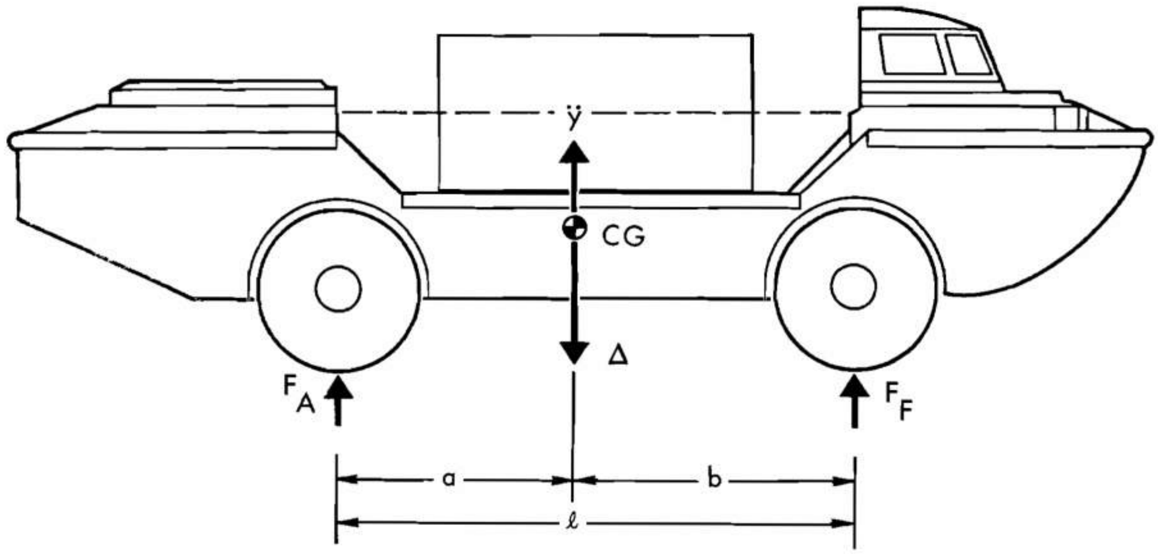


Figure 7-21. Forces for Maximum Bending Moment (Landborne)

$$M_{B_x} = F_F b_x - \sum_{i=1}^{x} w_i \ddot{y}_i \ell_i, \text{ ft-lb} \qquad (7\text{-}32)$$

where

M_{B_x} = bending moment at section x, ft-lb
b_x = arm between forward wheels and x, ft
w_i = weight of vehicle at i^{th} section, lb
$\ddot{y}_i$ = acceleration at i^{th} section, g's
ℓ_i = arm between i^{th} section and x, ft

A typical bending moment curve for the LARC V is shown in Fig. 7-22.

The bending moment applied to the hull when it is hoisted may be calculated in the same manner as for the land mode. Normally the accelerations are relatively small and the vehicles are handled without cargo aboard so that the forces are not critical. The proof test requirement for the lifting pads can be checked if there is any doubt about whether this may be a critical condition.

Another special loading condition that is sometimes critical for ships occurs during the launch when the bow is on the launching ways and the stern is afloat. There is a parallel for this in the amphibian cases when it is traversing the beach. Again, the forces should be no higher than when the vehicle is in the land mode.

7-3.1.2 Torsional Forces

7-3.1.2.1 Waterborne

Major torsional loads occur in the waterborne mode when running oblique to the waves where the bow on one side is supported on a wave crest and the stern of the other side is on the next crest. For ships, the torque is estimated by the following formula from Ref. 24 for full bodied vessels:

$$\text{Torque} = 0.22\, LWL\, B^3, \text{ ft-lb} \qquad (7\text{-}33)$$

LWL = waterline length, ft
B = beam, ft .

7-3.1.2.2 Landborne

For the land mode the maximum torque is imposed on the hull when wheels on opposite corners are lifted and the other two wheels are unloaded. This occurs simultaneously with an acceleration which increases the torque.

$$\text{Torque} \approx \frac{T}{4} \ddot{y}\, \Delta \frac{(\ell - b)}{\ell}, \text{ ft-lb} \qquad (7\text{-}34)$$

T = transverse distance from center of gravity to wheel, ft
$\ddot{y}$ = design acceleration at the center of gravity, g's

Δ = total weight of vehicle and payload, lb
b = longitudinal distance from wheel to center of gravity, ft
ℓ = wheelbase, ft

The torque forces from the land mode are substantially greater than those from the water mode unless the beam is extreme. The design torque then would be the higher torque value using the above equation assuming the design value of the acceleration.

7-3.1.3 Local Forces

Local forces are considered to be those that are applied in a sufficiently small area that they do not produce a significant stress in the hull as a unit. For amphibians these loads are the critical ones for most of the structure. In some cases, however, the stresses from the local loads must be combined with other stresses to determine those that are critical.

These loads can be applied by the water, suspension system, vibration, motion of the vehicle, collision with extraneous objects, and by a variety of special modes of operation such as towing.

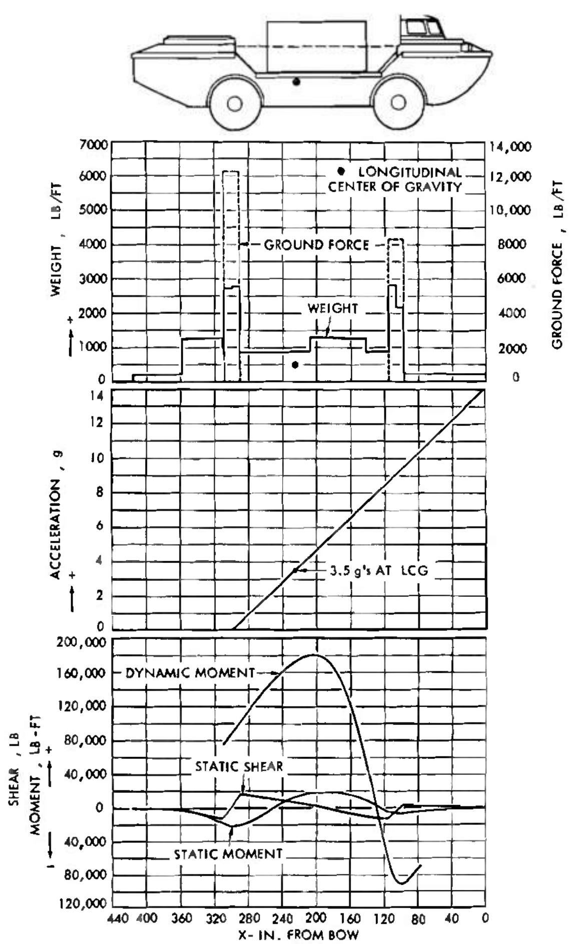


Figure 7-22. Bending Moment Curves for LARC V (Landborne)

7-3.1.3.1 Water Forces

The shell and its supporting structure will be loaded hydrostatically when afloat. For general use, the pressure is:

$$P_o = \rho g h, \text{ lb/ft}^2 \qquad (7\text{-}35)$$

where

P_o = pressure normal to the shell at the given depth h, lb/ft^2
ρg = 64 for salt water, lb/ft^3
ρg = 62.2 for fresh water, lb/ft^3
h = head of water, ft

The head of water will vary with the location of the hull, mission, and other characteristics. Normally for the smaller vehicles the smallest head should be to a point 4 ft above the shear line, as per Refs. 24 and 25. This minimum requirement is selected from experience on other vessels to provide loads for the normal conditions. There are other hydrodynamic loads that must be specifically considered, including wave slap on the vertical surfaces, slamming on the bottom, and breaking waves on the bow and stern.

7-3.1.3.2 Other Impact or Shock Forces

Most of the critical forces are generated from dynamic conditions. The previous paragraph discusses the local water loads when the vehicle is considered to be motionless. This paragraph covers the forces occurring when the amphibian collides with stationary objects and the local forces imposed from the suspension system in the land mode.

When alongside a ship or other essentially motionless object the amphibian will, at times, be pushed against the ship by the wave action. When the two are side by side, there is only a small impact because the water between them

acts as a cushion. However, there is a problem when the two are not parallel, as during the approach and departure of the amphibian from alongside a ship. The corners of the bow and particularly the quarters or corners of the transom are frequently damaged on both amphibians and landing craft. A solution to the bow damage problem is, where feasible, to round off the corners. The ideal shape is similar to that used for tugboats. If this is not feasible, then the structure in these areas must be designed to absorb the load.

The curve in Fig. 7-23 shows the relationship between the water surface velocity and the wave height for solitary waves of minimum stable length.

By use of the curve from Fig. 7-23 the maximum energy may be estimated from

$$E = \frac{V^2 \Delta}{2g}, \text{ ft-lb} \qquad (7\text{-}36)$$

where

E = approximate energy in the system due to wave action, ft-lb
V = speed of vehicle, ft/sec
Δ = displacement of vehicle, lb
g = 32.2 ft/sec^2

The preceding calculation neglects two phenomena. One is the virtual mass or assumed entrained water that acts with the vehicle to increase the energy level. The other is that the actual velocity for a floating object propelled by a wave is a function of the characteristics of the wave, hull form, and relative orientation of the body. For a heavily loaded boat shape such as an amphibian, the actual velocity will be less than the maximum surface water speed and this will reduce the maximum energy in the system.

Another local load that has led to extensive damage in the past has been loading cargo. This is particularly a problem in the higher sea states when receiving cargo from a ship. During that operation the amphibian is moving vertically with the waves, the ship is rolling, and the cargo is lowered using manual control. If the ship is rolling more than the operator can compensate, it can be assumed that the loading operation will be stopped. In general, cargo is lowered at a normal speed of about 25 ft/sec until it is near the amphibian and then it will be lowered slowly until it is down. The winchman, however,

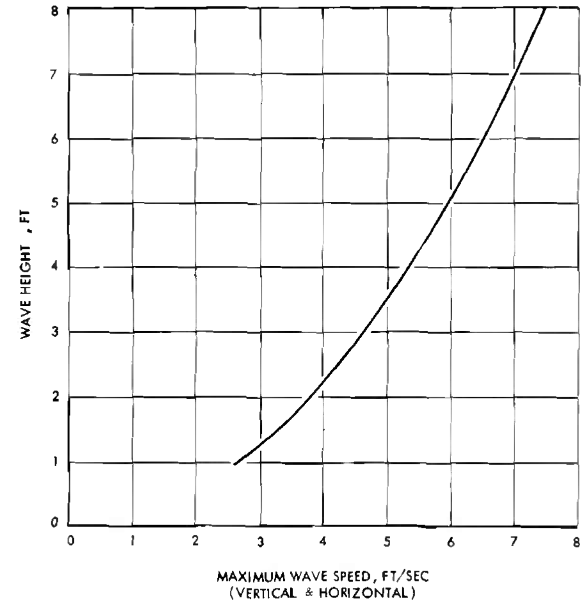


Figure 7-23. Wave Height and Maximum Speed for Solitary Deep Water Waves

usually cannot see over the side and he lowers on signal from the gangway man. Therefore, it is anticipated that on occasion the cargo will land with a relative motion of 25 ft/sec, plus the vertical speed from Fig. 7-23. In some instances the cargo may be dropped due to errors in judgment. Since a CONEX container may be loaded to a gross weight of 10,000 lb, it would, in general, be impractical to size the structure to withstand such a load.

Another typical type of force that cannot be countered in all cases is collision when operating on land. Since there is a great variety of sizes and shapes of objects that can be hit, it is generally not practical to calculate the possible loads unless there is a special hazard that will be encountered frequently for a particular design.

Another consideration of importance is the impact load to which the vehicle will be subjected through the suspension. Loads of this type come from shocks imparted to the wheels when they encounter obstacles, and shocks occurring when the vehicle is being transported.

Inputs with typical off-road obstacles are of such short duration that the most practical means of calculating them is using momentum equations. The momentum equations described the impact by relating linear impulse (force X Δ time) to changes of momentum- (mass X velocity), where "Δ time" is the time duration over

which the force acts. These equations and the calculations used to determine the forces are described in Ref. 26.

7-3.1.3.3 Semi-static Forces

There are a number of local loads that can be considered essentially as static loads.

The lifting pads are subjected to loads as shown on Fig. 8-2. If the specifications do not indicate these loads, the pads should be designed for a dynamic load factor of 3. If the pads are not centered about the center of gravity of the vehicle, they will not be equally loaded. Normally amphibians are hoisted without cargo but they may be hoisted in a sinking condition with a large amount of entrained water.

There are two philosophies with regard to the cargo lashings. One is that the cargo will be secured so that it will not come loose, and the other is that the cargo will break loose and fall overboard when the roll exceeds some value (less than the range of stability). In either case it is likely that the lashing pads on the amphibian will be built to retain the cargo in any eventuality. Forces on the lashings are determined from vector analysis for maximum roll angle without acceleration and maximum horizontal acceleration from rolling at zero roll angle. The procedure which follows may be used to determine the acceleration.

The natural period of roll is estimated from

$$T \approx \frac{0.44B}{\sqrt{GM}}\text{, sec} \tag{7-37}$$

where

T = natural roll period, sec
B = beam, ft
GM = metacentric height, ft

If we assume sinusoidal motions, the maximum roll acceleration is

$$\ddot{x} = \frac{0.689\,\ell\theta_M}{T^2} \tag{7-38}$$

where

ℓ = vertical arm from CG to point of interest, ft
θ_M = maximum roll angle to one side, deg
$\ddot{x}$ = maximum horizontal acceleration from rolling, ft/sec^2

and eliminating T^2 from the Eq. 7-38

$$\ddot{x} = \frac{3.55\,\ell\theta_M(GM)}{B^2} \tag{7-39}$$

This equation may be used for $\theta_M < 15°$. At greater roll angles, the equation gives values that are too high. In general, it will be found that the static loads at the maximum roll angles are higher than the acceleration loads. These loads are not additive since they are out of phase. The wheel loads from wheeled cargo such as howitzers can also be significant for the design of the cargo deck. These loads can be determined using Ref. 27 (which lists the load per axle) and the design maximum vertical acceleration $\ddot{y}$:

$$\text{Max wheel load} = \frac{(\ddot{y}+1)\text{ axle load}}{\text{Number of wheels on the axle}} \tag{7-40}$$

Ref. 26 shows the assumed area and pressure distribution for typical tires.

Pad eyes, or equivalent, are normally fitted for towing fixtures which will permit the amphibian to be towed both on land and in the water. The fixtures for use while waterborne must be above the waterline and, preferably, on the deck. The loads to be transmitted are:

a. For towing cleats (waterborne) the strength of the strongest line that will fit. Nylon lines are normally used for mooring purposes.

b. For towing pads (land) the strength of the standard towline.

Both should have a factor of safety of 3 on the ultimate strength of the material (Ref. 28).

The suspension system is subject to impact loads which are the controlling factor in its design.

Some method of changing tires is required. Hard points for jacking near each wheel should be available and clearly marked for that purpose. Each of these points should be designed to support half the vehicle weight.

7-3.1.3.4 Vibration

Vibration of structural members in an amphibian can originate from the engine, propulsor, drive train, suspension system, or the action of waves in a seaway. Since the power is usually high relative to a conventional truck, it is possible to generate significant vibrations which

can add to the stress in some of the members.

There are certain frequencies, such as the engine operating rpm or the blade rate of the propeller, that will occur throughout the life of the vehicle. It is possible to calculate with a good degree of accuracy most of the natural frequencies of the structure. The calculations are tedious and time-consuming so that usually they are neglected and the prototype vehicle is closely checked for vibrations. Undesirable vibrations are eliminated where possible or reduced by the addition of extra stiffeners. Where the suspension system of wheeled cargo is in synchronization, it can be eliminated with proper tie-downs.

7-3.2 MATERIALS

The selection of materials for the construction of the hull is a very important phase of the structural design, and can play an important part in the life-cycle cost, reliability, and maintainability of the vehicle.

Important considerations for the selection of materials include:

a. Availability of the necessary shapes in the time frame required

b. Cost of the basic material and cost to fabricate

c. Weight of the completed hull

d. Repairability in the field

e. Maintenance required, particularly in the sea water environment

f. Strength and size of framing (large frames reduce the mobility of maintenance personnel)

g. Familiarity of the constructing facility with the material

This list is not all-inclusive and there are many interrelationships among the items listed; they are, however, major items that must be considered prior to selection of a material.

The materials available at this time that could be considered are aluminum, steel, and Fiberglas. The specifications for these materials as used by the Navy are listed in Table 7-3.

7-3.2.1 Comparison Parameters

There are a number of parameters that can be used to compare materials without specific reference to a design. These are useful to eliminate potential construction materials prior to designing the structure. Absolute comparisons for any case require preparation of complete optimized designs in each material.

The base cost of the materials is shown in Table 7-4. A variety of special charges or discounts are applied to these costs but the relative final costs will be close to those shown.

Table 7-5 shows approximate relative costs of fabricated aluminum materials for commercial structures similar to the LARC V or LARC XV in quantities of 20 or more. These are only an indication of the trends with respect to aluminum because reliable information is not available from manufacturers equally experienced in using all materials.

The strength-weight ratio, i.e., tensile strength/density, is a gross indication of the relative weight of a structural material. Table 7-6 lists the comparisons of interest.

Ductility is important as an indication of resistance to cracking at stress concentrations. Table 7-7 lists elongation values, a measure of ductility.

The ability to absorb impact with water (as in slamming) or solid objects may be judged by the area under the stress-strain curve, which is an indication of the energy that can be absorbed. The distance along the strain axis depends on the design criteria. Table 7-8 lists the areas of interest.
These values do not consider weight, but are based on an equal volume. Table 7-9 lists the approximate values on an equal weight basis.

Deflection is another criterion for design in some instances. Table 7-10 lists the approximate relative deflections.

Perhaps the most important consideration for the material selection is the fatigue strength. This varies for each material, and for each material it varies primarily with the stress history. Time, environment, and other factors have a major effect in some instances, but not normally for use in amphibians.

The stress history is the order and magnitude of each stress which the structure sustains. From the available data, it appears that the order of the application of stresses is of secondary importance compared with the number and magnitude in the applied stress spectrum. It is at this particular point in the analysis that engineering estimates must start since data on fatigue strength for various applied stress spectra are not available. There is, however, a sufficient quantity of data for the unique case where the

TABLE 7-3 MILITARY SPECIFICATION NUMBERS FOR HULL MATERIALS (Ref. 28)

Bars and shapes

Steel, medium	MIL-S-20166, Grade M
Steel, high tensile	MIL-S-20166, Grade HT
Steel, high yield strength (extruded)	MIL-S-22664
Steel, high yield strength (rolled)	MIL-S-22958
Steel, high yield strength (bars)	MIL-S-21952

Plates

Steel, medium	MIL-S-22698
Steel, high tensile (surface ships)	MIL-S-16113, Grade HT
Steel, high tensile (surface ships or submarines)	MIL-S-24094
Steel, high yield strength	MIL-S-16216
Steel, special treatment	MIL-S-20154

Tubing

Tubing, steel	MIL-T-16343, Type 1

Reinforcements such as doublers, reinforcing rings, and flat bar reinforcement of stiffeners, for members designated to be high tensile steel or high yield strength steel, shall be of the same material. Where approved the following may be used:

Iron castings, extra tough, MIL-I-22243.
Iron castings, gray, Fed. Spec. QQ-I-652 or ASTM Standards A48 to A278.
Malleable iron castings, Fed. Spec. QQ-I-666, or ASTM Standard A47 for minor installations.
Ductile iron, MIL-I-17166, for minor installations.
Marine boiler plate, Fed. Spec. QQ-S-691, Class B, for parts that require a large amount of working under repeated heatings, such as for flanging.
Steel castings, high yield strength, MIL-S-23008.
Steel forgings, high yield strength, MIL-S-23009.

Use	Aluminum Alloy Spec.	Aluminum Alloy Alloy	Condition Grade or Class	Remarks
Castings	QQ-A-601	356	3M	Note 1
Castings	QQ-A-601	195	4M	Note 2
Castings	QQ-A-601	214	5M	Note 3
Plates	QQ-A-250/7	5086	H32	Note 4
Plates	QQ-A-250/9	5456	H321	Note 5
Plates	QQ-A-250/10	5454	H34	Note 6
Plates	QQ-A-250/11	6061	T6	Note 7
Shapes				
Extruded or rolled	QQ-A-200/5	5086	H111	Note 4
Extruded	QQ-A-200/7	5456	H111	Note 5
Extruded or rolled	QQ-A-200/6	5454	H111	Note 6
Rolled or drawn	QQ-A-225/8	6061	T6	Note 7
Extruded	QQ-A-200/8 or ASTM B221	6061	T6	Note 7

(cont'd)

TABLE 7-3 (Concluded)

Use	Aluminum Alloy Spec.	Aluminum Alloy Alloy	Condition Grade or Class	Remarks
Tubing				
*Drawn or extruded	WW-T-700/5	5086	H32	Note 4
**Welded	QQ-A-250/7	5086	H32	Note 4
All sizes	WW-T-700/6	6061	T6	Note 7

*Drawn preferred. Consideration must be given to wider dimensional tolerances when extruded is used.

**Tubing manufactured from plate and having a longitudinal welded seam may be used for tubing sizes that are not available in drawn tubing.

Note 1 - For use with complex castings where castability, pressure tightness, strength, and resistance to corrosion are required. Will respond to heat treatment to improve strength. For applications requiring high casting quality and excellent fluidity.

Note 2 - High tensile, with less corrosion resistance than Class 1, 3, 5, 7, and 8. Heat treatment is required. For uses such as ammunition stowages, frames and sills for joiner doors, and ladder treads.

Note 3 - For use wherever good tensile strength and relatively high resistance to corrosion are required. Heat treatment is not required. For applications similar to Class 4 but requiring resistance to corrosion at a sacrifice of tensile properties.

Note 4 - Shall be used for applications where higher strength is not required.

Note 5 - Shall be used for applications where high strength is required and where higher cost is warranted.

Note 6 - For a structure subject to elevated temperatures over 150°F such as the upper portion of smokestacks.

Note 7 - This alloy shall be used for nonwelded structure only.

Fiberglas	MIL-P-17549
Polyester Resin	MIL-R-7575B
Polyester Resin	MIL-R-21607
Woven Roving	MIL-C-19663A
Mat	MIL-M-15617A

TABLE 7-4 BASE COST OF HULL MATERIALS

Material	¼ in. Plate $/lb	¼ in. Plate $/ft²	2 in. X 3 in. X ¼ in. Angle $/lb	2 in. X 3 in. X ¼ in. Angle $/ft²
Mild Steel	0.09	0.92	0.10	0.41
High Tensile Steel	0.24	2.45	0.25	1.03
Aluminum (5456-H32)	0.46	1.59	0.74	1.06
Fiberglas Cloth	0.37	0.64*	0.37	0.26*
Polyester Resin	0.49		0.49	

*Assuming 50% glass and 50% resin by weight (sp. gr. 1.15)

TABLE 7-5 RELATIVE FABRICATION COSTS PER POUND (WITH RESPECT TO ALUMINUM)

Material	*Relative Cost, %*
Mild Steel	30
High Tensile Steel	45
Fiberglas	49

TABLE 7-6 STRENGTH-WEIGHT RATIOS

Material	*Yield Strength,* lb/in.^{3}	*Ultimate Strength,* lb/in.^{3}
Steel	115	230
Fiberglas	(no yield)	240 - 520
Aluminum (5456-H24)	406	530

TABLE 7-7 TYPICAL MINIMUM ELONGATION

Material	*Elongation in 2 in., %*
Steel	24
Fiberglas	2.5
Aluminum (5456-H24)	9

TABLE 7-8 RELATIVE AREAS UNDER TYPICAL STRESS-STRAIN CURVES

Material	*Area to Yield Stress*	*Area to Ultimate Stress*
Steel	1	2½
Fiberglas	(no yield)	1
Aluminum (5083 welded)	4	3

TABLE 7-9 RELATIVE ENERGY ABSORBED FOR EQUAL WEIGHTS

Material	*Energy to Yield Stress*	*Energy to Ultimate Stress*
Steel	1	1
Fiberglas	(no yield)	1.8
Aluminum (5083 welded)	13	3.2

applied stress is constant. Fig. 7-24 shows the standard type of fatigue S-N curves for steel, aluminum, and Fiberglas. It is obvious from these curves that great care must be exercised in the estimate of the number of cycles selected for design.

The bottom structure is stressed primarily at each wave, and the cargo well deck will also be highly stressed at each wave when cargo is aboard. The sides may only be severely stressed when coming alongside. Consequently, the fatigue strength will vary depending on the location of the structure. Assuming 10 yr of active service with an average use underway of 20 hr per week and a stress cycle averaging 1 cps, then the number of cycles is 3.7×10^7 cycles.

By use of Fig. 7-24, Table 7-11 was developed to show the influence of fatigue stress at 3.7×10^7 cycles.

Table 7-12 lists the mechanical properties for various materials. The data are from the listed sources.

7-3.2.2 Steel

Steel is the basic construction material for practically all automotive vehicles and ships. Its use is limited in aerospace due to weight and in small boats because of excessive corrosion problems. The reasons for its extensive use include low material cost, low fabrication cost, high mechanical characteristics, excellent availability, and practically universal familiarity with construction and repair procedures.

Two basic types of steel for construction have been developed to include a consistently high grade of material to withstand the dynamic loading that is prevalent. The primary one is a

TABLE 7-10 RELATIVE DEFLECTIONS

Material	*Tensile Modulus of Elasticity*	*Equal Volume*	*Equal Weight*	
			Beam in Tension	*Beam in Bending*
Steel	29×10^6 psi	1	9	9
Fiberglas	1.4×10^6 psi	21	4½	2
Aluminum	10.3×10^6 psi	3	1	1

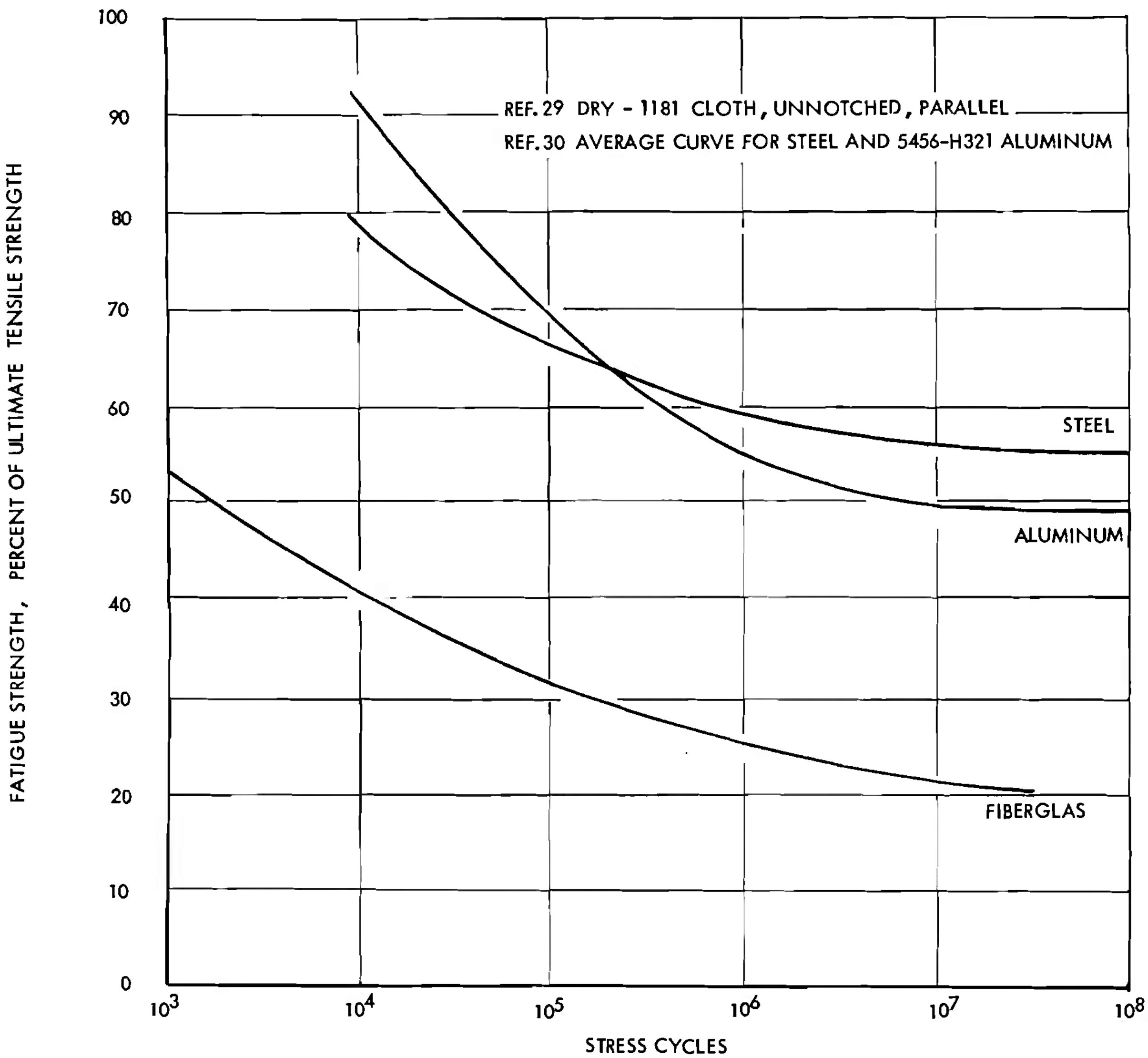


Figure 7-24. Fatigue Strength as a Function of Stress Cycles

TABLE 7-11 RELATIVE FATIGUE STRESS/DENSITY

Material	3.7×10^7 *Cycle Fatigue Stress,* psi	*Relative Values of Fatigue Stress/Density*
Steel ·	37,000	1.3
Fiberglas	7,000	1.0
Aluminum (5456-H34)	21,000	1.9

low carbon killed or semi-killed steel manufactured with rigid controls. The Navy classifies this as medium steel, and it conforms to MIL-S-22698 for plates and MIL-S-20166, Grade M for bars and shapes.

The second type is a low-alloy, high tensile strength steel generally labeled HTS. These meet the following specifications: ASTM A-242, MIL-S-20166, Grade HT for bars and shapes, and MIL-S-16113, Grade HT or MIL-S-24094 for plates.

As a construction material for amphibians, steel has many advantages, including:

a. Excellent availability

b. Low cost, even for HTS

c. Properties are well known

d. Strength is high except at very low temperatures

e. Fireproof

The disadvantages are:

a. Corrosion in a marine environment

b. Heavy weight

Since, in general, only thin plates (10 and 12 gage for the LARC LX) are required for the shell to obtain adequate strength, corrosion must be minimized. There are coatings available that can practically eliminate the problem. These coatings have a long life and are expensive.

The problem of weight is not so easily solved. In fact, a steel hull will weigh about twice as much as a comparable aluminum hull. This one disadvantage essentially eliminates steel from serious consideration as a construction material except when the design penalty for added weight is low, such as in a swimming type vehicle, i.e., low water performance.

7-3.2.3 Fiberglas

Fibrous glass reinforced plastics, commonly called "Fiberglas" in this country, is a very successful material for boats. In the smaller boat sizes it dominates the industry. This material is relatively new and its technology is still moving forward at a fast pace. The design information of today (1969) may well be out of date in a few years.

Fiberglas is a combination of high tensile strength glass filaments in a plastic structure. The glass fibers carry the load and the plastic holds the fibers in position and distributes the forces in much the same way as steel reinforced concrete functions. The glass fibers are available in a wide range of sizes and configurations which vary from woven material to glass mat which is a layer of short fibers in a random pattern. There is also a variety of plastic resins available, primarily the polyesters and epoxies. The possible combinations of different configurations of glass fibers and resins are very useful in optimizing the use of the material for the particular structure needed. The cost of fabrication is low if there is a sufficient number of hulls made to reduce the cost of the molds per hull.

Other important characteristics are (1) repairability is good if the working temperature is above about 50° F, and maintenance is relatively low since it is an inert material that does not oxidize readily; (2) the weight of material required is higher than aluminum and lower than steel; (3) if the structure is designed to a deflection criterion, then a sandwich-type configuration can be used which will be light; and (4) many shops now specialize in Fiberglas construction, and there are personnel available that are well trained in using the material. In general, however, these special facilities are not in the companies that are likely to build amphibians.

TABLE 7-12 MECHANICAL PROPERTIES OF CONSTRUCTION MATERIALS

Material	Steel Mild	Steel Hi Tensile	Fiberglas see note (1)	Aluminum 5456-H321 (2) Parent	Aluminum 5456-H321 (2) HAZ (3)	Tempered Plate Glass
Tensile Strength, ksi (7)						
Ultimate						
Minimum		70	9.8	46	42	25
Typical	67		33	51		30
Yield						
Minimum		50		33	26	
Typical	33			37		
Elongation						
Minimum		18 (6) 22 (5)		12 (5)		
Typical	25 (5)			16 (5)		
Compression Strength, ksi						
Yield						
Minimum			4.2	32	25	25
Typical	67		15.8	30		30
Shear Strength, ksi						
Ultimate						
Minimum				27	25	
Typical	50		11			
Yield						
Minimum				19	15	
Bearing Strength, ksi						
Ultimate						
Minimum				84		
Yield						
Minimum				56		
Weight, lb/in.^{3}	0.283	0.283	0.059	0.096		0.091
Specific Gravity	7.84	7.84	1.64	2.66		2.53
Modulus of Elasticity, ksi						
Tension/Compression	30,000	30,000	2060/1900	10,300		10,000
Shear				3,850		
Poisson's ratio	0.26	0.26	.14/.65/.11 (4)	0.33		0.23
Coefficient of Expansion, °F^{-1}	0.65×10^{-5}	0.65×10^{-5}		1.33×10^{-5}		0.49×10^{-5}
Refs.	28	50	29	31		51

Notes:

1. 25-27 oz woven roving with polyester resin 1/4 in. thick, 51 percent glass by weight, immersed in water at room temperature for 30 days.
2. 1/4 in. thick sheet
3. HAZ - Heat affected zone
4. 0° to weave/45° to weave/ and 90° to weave
5. Percent in 2 in.
6. Percent in 8 in.
7. ksi - 1,000 lb/in.^{2}

From Table 7-12 it can be seen that the strength of Fiberglas is high and the configuration can be varied throughout the structure to take full advantage of this strength. There are, however, four areas that must be carefully considered in Fiberglas design:

1. The peel strength of Fiberglas is low. This is particularly important where bulkheads or other structures are added perpendicular to the hull with bonding angles.

2. The laminate bearing strength is also low and results in local crushing from mechanical fasteners unless the bearing surface is enlarged over that available with normal size washers. Consequently, in an amphibian the problems of attaching the components of the machinery and drive trains must be specially considered for Fiberglas to be a successful construction material.

3. Besides these problems of attachment, the impact strength is low, as shown in Table 7-9.

4. The flexibility is high which can cause alignment problems.

7-3.2.4 Aluminum

Of the three potential construction materials available, aluminum construction will result in the lightest hulls. Aluminum has been successfully used for the LARC V and LARC XV vehicles. Although there are problems, it is probable that future amphibians of 5- to 15-ton size will be constructed of aluminum. For this reason the characteristics of aluminum are presented here in more detail.

Two types of wrought aluminum alloys are available for structural purposes—heat treatable and nonheat-treatable. The heat treatable alloys gain a major portion of their strength by a post fabrication heat treatment. If the structure is too large to fit in an oven, either the lower strength must be accepted in the way of welding and the other areas where heat is applied during manufacturing or mechanical fasteners and cold forming may be used. The nonheat-treatable alloys use a combination of alloying elements and cold work to achieve high strength.

The wrought alloys are designated using a four-digit system. The first digit indicates the alloy group, and these are defined in Ref. 31 and in other references. The "marine" alloys are in the 5000 series which has magnesium as the main alloying element. The ultimate tensile strength ranges from 63,000 psi to 31,000 psi varying with the alloy and the temper. The temper designation system is used for all forms of wrought and cast aluminum alloys except ingot and is based on the sequences of basic treatments used to produce the various tempers. The temper designation follows the alloy designation with the basic identifier consisting of one letter, and variations are indicated by a series of digits. Table 7-13 lists the basic temper designations.

Two special designations of interest are:

a. H311—Alloys with over 4 percent magnesium which are strain hardened less than the amount required for a controlled H31 temper.

b. H321—Alloys with over 4 percent magnesium which are strain-hardened less than the amount required for a controlled H32 temper.

Other properties of the 5000 series are:

a. Excellent corrosion resistance in the marine environment

b. Easily weldable and initial mechanical properties are not greatly affected by welding

c. Good formability

d. Good availability

The mechanical properties of the two higher strength most common alloys in the 5000 series are listed in Table 7-12.

Standard and maximum plate sizes for the more popular 5000 series alloys and for 6061-T6, the heat treatable alloy most commonly used in the marine structures, are listed in Table 7-14. Standard structural shapes are stocked at local warehouses throughout the country mainly in 25-ft lengths. Other sizes are frequently available locally. to suit the special requirements of the area. Mill orders of extrusions can be supplied in any alloy with the length limited only by the available transportation. Aluminum is generally formed cold using the same equipment and techniques used for steel. Many members which would require hot forming in steel can be formed cold in aluminum.

Various aspects of cold forming aluminum can be found in Ref. 32. In addition, Table 7-15 and Figs. 7-25 to 7-32 are presented for use in designing bends in developable surfaces. The major difference between cold forming aluminum and steel is the greater spring-back of the aluminum. Where necessary, hot forming can

TABLE 7-13 BASIC TEMPER DESIGNATIONS

Designation	*Description*
F	As fabricated
O	Annealed
H	Strain hardened (nonheat-treatable alloys)
W	Solution heat treated
T	Thermally treated (heat treatable alloys)

The first digit of the subdivisions are:

1 Strain hardened only

2 Strain hardened and partially annealed

3 Strain hardened and stabilized. A first digit of 3 is used only for the alloys with significant amounts of magnesium because they will gradually age-soften at room temperature unless they are stabilized.

The second digit indicates the degree of strain-hardening remaining after stabilization. Where

0	Annealed
1, 2, 3, 4, 5, 6, 7	Ultimate strength approximately equal to the annealed ultimate plus the number/8 of the difference between the ultimate strength of the annealed and of the 75 percent cold reduction condition
8	75 percent cold reduction

be used with aluminum where the temperatures are limited to about 400° F for a short period of time. This temperature must be carefully considered for each alloy. Unlike steel, there is little or no visual change in the appearance of aluminum at reasonable forming temperatures so that great care must be exercised. Cutting of aluminum in shipyards is normally done by sawing or shearing. Normal wood working portable tools with special blades are very useful. High-speed arc cutting is feasible and routers are often used for intricate shapes or where a number of identical pieces can be cut from one pattern.

Leaks almost always develop at riveted connections after the structure has been stressed; accordingly, riveting is impractical for fastening the shell. Rivets or other mechanical fasteners can be used where there are no requirements for a watertight joint.

Welding is now the primary means of fastening the structural sections together. Two main types of welding guns are in use. TIG, called "tig", for tungsten-arc, inert-gas shield; and MIG, called "mig", for metal-arc, inert-gas shield. The TIG system has a tungsten electrode which forms the arc, and a hand-held welding rod. A jet of inert gas, usually a mixture of

TABLE 7-14 ALUMINUM PLATE PRODUCTS — STANDARD AND MAXIMUM SIZES AVAILABLE (Ref. 52)

(All Dimensions in in.)

	Unit[1]	Standard Sizes				Maximum Sizes			
Thickness	Wt, lb/ft²	5052-H32	5086-H32	5456-H321	6061-T651	5052-H32	5086-H32	5456-H321	6061-T651
1/4	3.46	48 x 144	48 x 144 72 x 144	60 x 144	36 x 96 48 x 144 60 x 144 72 x 144	120 x 480	132 x 480	120 x 480	120 x 480
5/16	4.31	48 x 144	48 x 144 72 x 144		36 x 96 48 x 144	120 x 480	132 x 480	120 x 480	120 x 480
3/8	5.2	48 x 144	48 x 144 72 x 144	60 x 144	36 x 96 48 x 144 60 x 144	120 x 480	132 x 480	120 x 480	120 x 480
1/2	6.9	48 x 144	48 x 144	60 x 144	36 x 96 48 x 144	120 x 540	132 x 540	120 x 480	120 x 480
3/4	10.4	48 x 144	48 x 144	60 x 144	36 x 96 48 x 144	125 x 540	120 x 540	120 x 480	132 x 540
1	13.8	36 x 96		60 x 144	36 x 96 48 x 144	96 x 540	96 x 540	120 x 480	132 x 540

[1]Weights given above apply to 5052, 5086 and 5456. These must be multiplied by 1.02 to apply to 6061.

		C102 (Diamond) Pattern Tread Plate	
1/8	2.0	48 x 192; 60 x 192; 48 x 144	60 x 300
3/16	3.0	48 x 192; 60 x 192	72 x 360
1/4	3.9	48 x 192; 60 x 192	72 x 360
3/8	5.6	48 x 192	60 x 360
1/2	7.3	....	60 x 360
		Abrasive Tread Plate	
1/8	1.7	48 x 144	48 x 144
3/16	2.6	48 x 144	48 x 144
1/4	3.4	48 x 144	48 x 144
3/8	5.1	....	48 x 96

TABLE 7-15 APPROXIMATE RADII FOR 90° COLD BEND SHEET AND PLATE

Al. Alloy and Temper	All Radii in In. (1)				
	1/8	3/16	1/4	3/8	1/2
5052-0	0	0-3/16	0-1/4	3/16-9/16	1/2-1
5052-H32	1/16-3/16	1/8-5/16	1/8-3/8	3/8-3/4	3/4 - 1-1/4
5052-H34	3/16-5/16	5/16-7/16	1/2-3/4	3/4 - 1-1/8	1-1/4 - 1-3/4
5086-O	0-1/8	0-3/16	1/8-1/4	3/8-9/16	1/2-1
5086-H32	1/8-1/4	5/16-3/8	3/8-5/8	3/4-15/16	1-1/4 - 1-1/2
5086-H34	3/16-5/16	3/8-9/16	1/2-3/4	15/16 - 1-5/16	1-1/2-2
5456-O	0-1/8	1/8-3/16	1/8-1/4	3/8-9/16	1/2-1
5456-H321	1/4-3/8	3/8-9/16	1/2-3/4	1-1/8 - 1-1/2	1-1/2-2
6061-0	0	0-3/16	0-1/4	3/16-3/4	1/2 - 1-1/4
6061-T6 (2)	1/8-1/4	5/16-9/16	1/2-1	15/16 - 1-1/2	1-1/2 - 2-1/2

(1) Minimum permissible radius over which sheet or plate may be bent varies with nature of forming operation, type of forming equipment, and design and conditions of tools. Minimum working radius for a given material or hardest alloy and temper for a given radius can be ascertained only by actual trial under contemplated conditions of fabrication.

(2) Tread Plate may require bend radii up to as much as twice the values listed for plain sheet and plate.

helium and argon, is fed by the welding gun around the electrode. TIG, with respect to MIG, requires more skill and generally is slower and produces more distortion. It is no longer used extensively for marine structures except for repair work.

The MIG system employs a gun with the welding wire used as the electrode, driven at various constant speeds. It also includes use of a jet of inert gas for shielding. These guns may be either air- or water-cooled. Outlines of typical MIG guns are shown in Fig. 7-33.

In general the weld quality required will govern the cleanliness required for welding. It is essential to degrease the prepared edges if lubricants are used in the preparation of the edges. Also the oxide film must be removed with a stainless steel wire brush if a high quality weld is required. These brushes should be used only for this purpose. A new film forms immediately after the original oxide film is removed and grows in thickness until stability is reached. The growth rate and eventual thickness of film depend on the environment. For each project it is necessary to determine the maximum allowable time between cleaning the weld area and welding. For one amphibian where the aluminum was chemically cleaned, a maximum allowable time of 24 hr was established.

Only two electrodes, alloys 5556 and 5356, are widely used in marine applications. Many others are available for special purposes. Alloy 5556 provides the highest weld strength. It is the only material recommended for use with 5456 and provides the highest fillet strengths with 5086 and 5052. Alloy 5356 provides high weld strength and is used with 5086 or 5052 parent metal. These two electrodes are also used when attaching 6061 structural shapes to a 5000 series plate.

Welding procedures are essentially the same as those used for steel except that a higher order of cleanliness is required, which should include the work area in the vicinity. The work must be shielded from the wind so that the gas shield is not blown away from the weld puddle. Intermittent welding can be used with continuous welds at the ends. The Navy requires double continuous welds in most cases to reduce cracks from shock loads. Welding in any position is practical although a better weld will generally be made in the down-hand position. Particular care must be used to allow room for the large snout on the gun, as shown on Fig. 7-33.

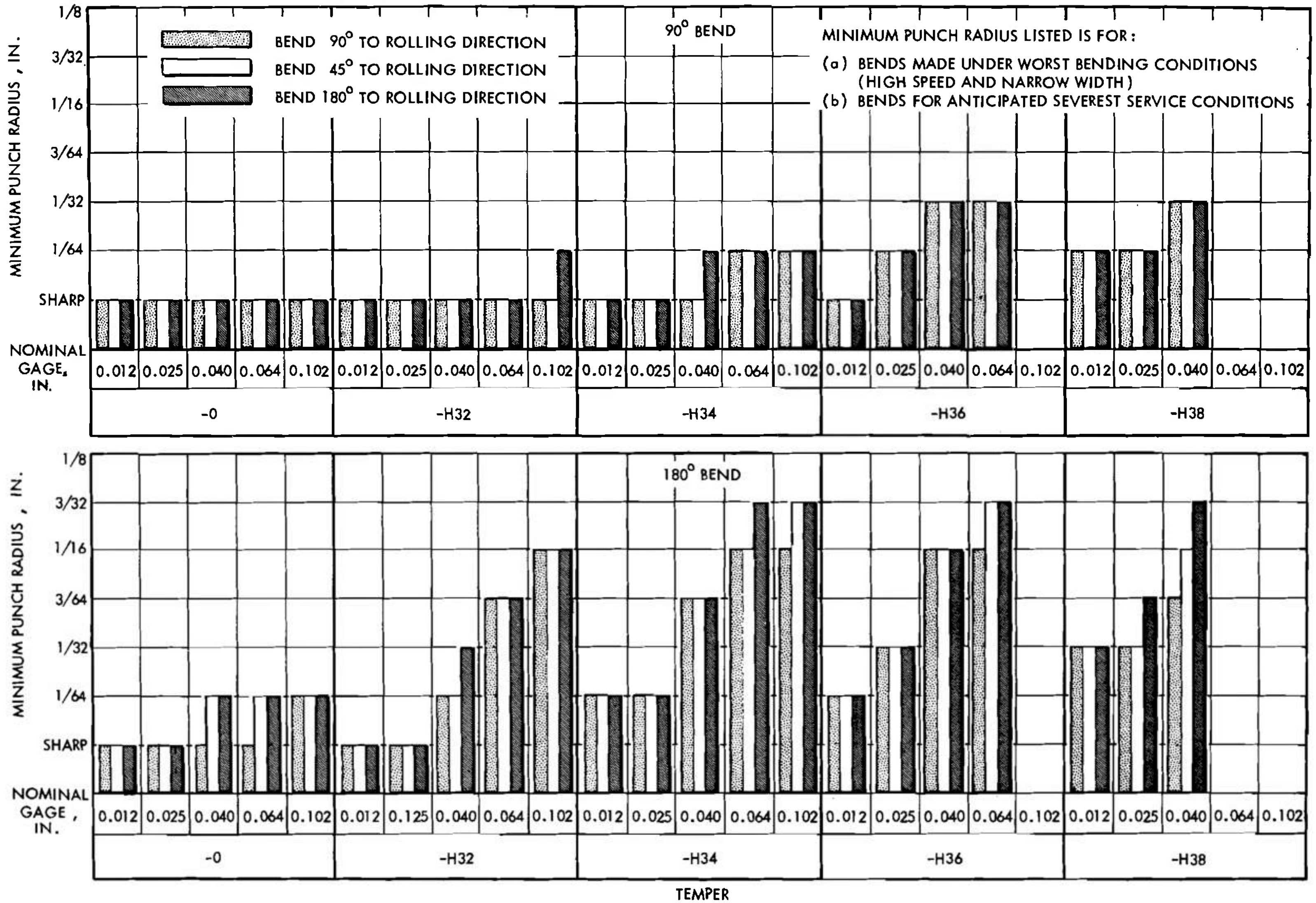


Figure 7-25. Bend Characteristics for Alloy 5457 Sheet

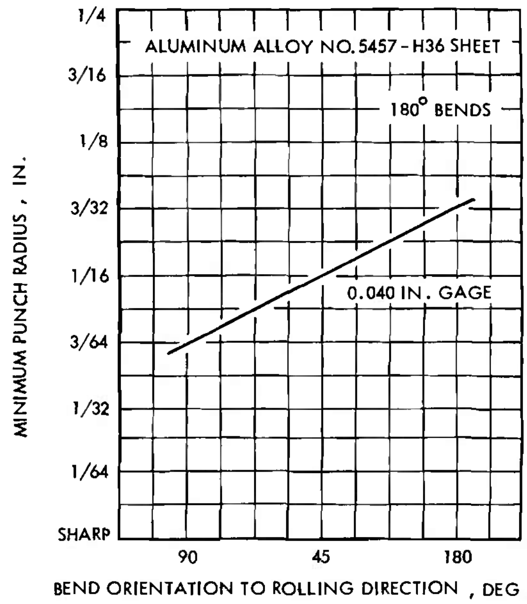


Figure 7-26. Effect of Bend Orientation on Bend Characteristics

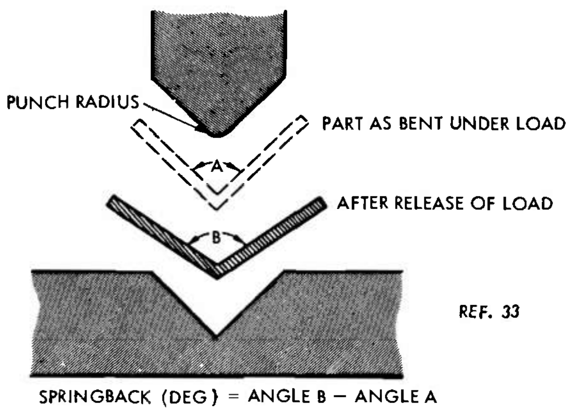


Figure 7-27. Springback Test Tooling

One of the advantages of aluminum with respect to steel is the high resistance to salt-water corrosion. This characteristic has been carefully investigated and proven by tests and in general practice. The marine alloys exhibit a self-limiting type of corrosion because the oxide formed is hard and tends to protect the metal underneath it. Stress corrosion is possible with the 5000 series. However, stress corrosion can be eliminated with proper combinations of alloying elements (magnesium limited to not more than 10 percent), temper, stress level, and the degree of cold work. For those situations where high stresses occur in material that has been extensively cold worked, the selection of the temper of the material used should be specially considered. The total cold-working that must be considered includes the original strain hardening done by the aluminum manufacturer and specified in the temper designation, and that done in cold-forming the plate or shape to fit by the vehicle manufacturer. The minimum bend radii for 90-deg cold bends given in Table 7-13 are, to a certain extent, limited by this consideration. Smaller radii should be hot formed. Service temperature should not exceed about 150°F without special consideration of the stress-corrosion problem. Where higher temperatures are unavoidable, such as in a stack enclosure, the 5454 alloy should be used.

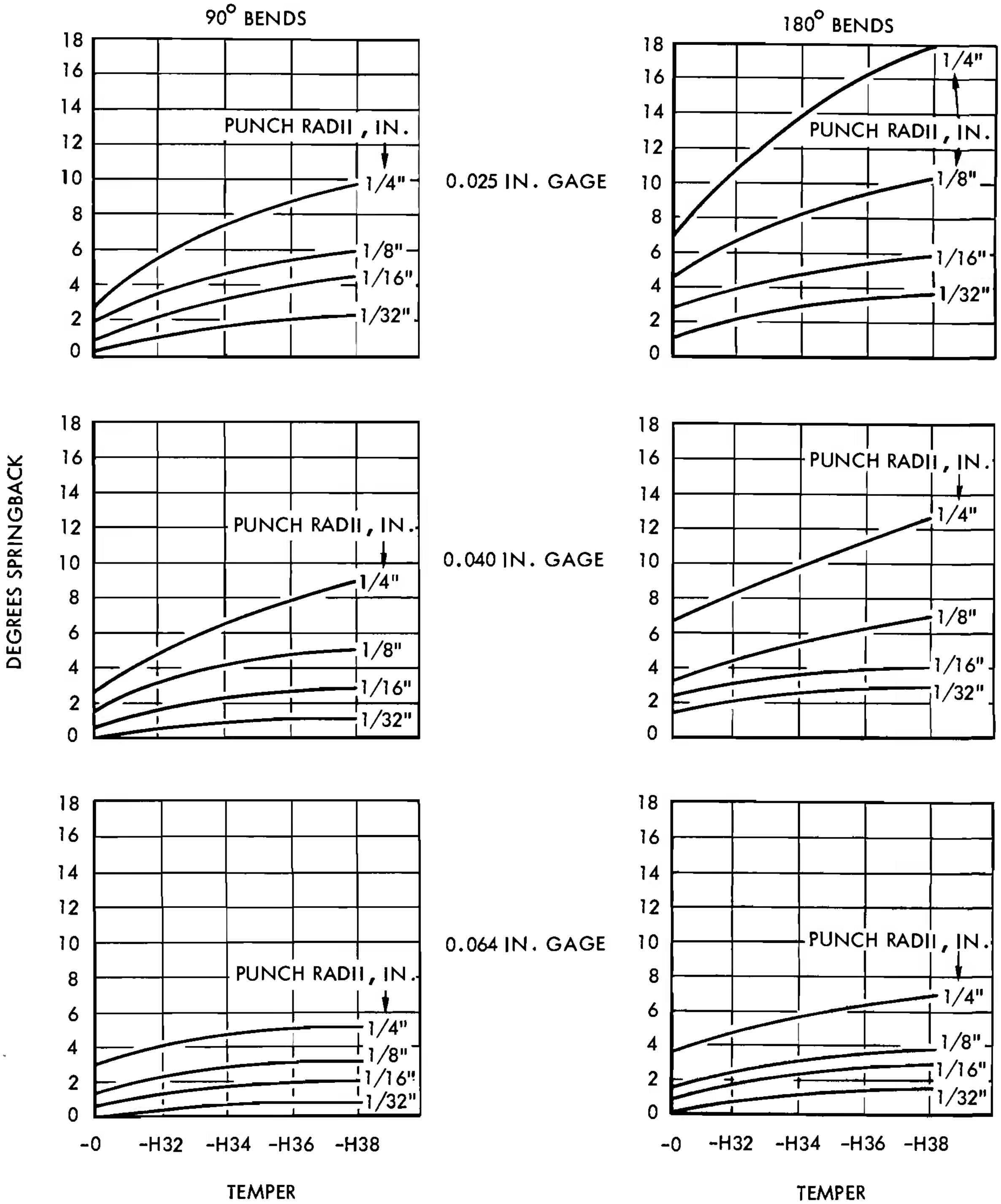
Galvanic corrosion resulting from the use of electrically connected dissimilar metals in an electrolyte such as salt water can be excessive in an aluminum hull. Table 7-16 lists the galvanic series where the more anodic metal is corroded. There are three general techniques for eliminating or reducing galvanic corrosion:

1. Alter the materials so they are the same or very close on the galvanic series. This may be done by placing a sacrificial anode in the vicinity. Magnesium alloys are frequently used for this purpose on aluminum hulls.

2. Eliminate the electrical path between the two electrodes, using any of a variety of caulking materials. The three most popular types are polysulfide, neoprene, or butyl-base materials which are available in either liquid or tape form. All electrical paths, including mechanical fasteners, must be eliminated and not allowed to reform during the life of the vehicle.

3. Isolate one or both materials from the electrolyte with a paint or other coating.

Crevice corrosion is a phenomenon that occurs in locations where the oxygen dissolved in sea water is depleted due to chemical action, and not replaced. Then if the passive film is broken in one spot a local galvanic cell is started and, since there is no oxygen, the passive film is not re-established. These conditions are prevalent where materials are joined by mechanical fasteners and the crevices are so small that there is no water flow. Crevice corrosion is generally not a problem for aluminum or copper alloys but it is for most of



ALL SPRINGBACK PLUS; OTHER RADII IN RELATIVE ORDER

SPRINGBACK FOR BENDS AT 90°, 45°, AND 180° TO ROLLING DIRECTION SIMILAR

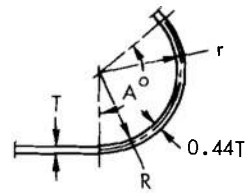
Figure 7-28. Springback of Alloy 5457 Sheet at Various Radii

FOR CALCULATIONS OF BEND ALLOWANCE WITH RADII GREATER THAN 1 IN. USE THE FOLLOWING FORMULA:

$0.017453\,rA$ = BEND ALLOWANCE

r = BEND RADIUS + 0.44 MATERIAL THICKNESS
A = NUMBER OF DEGREES IN BEND

CHANGE FRACTIONS OF DEGREES TO DECIMAL EQUIVALENT BEFORE SUBSTITUTING IN THE FORMULA



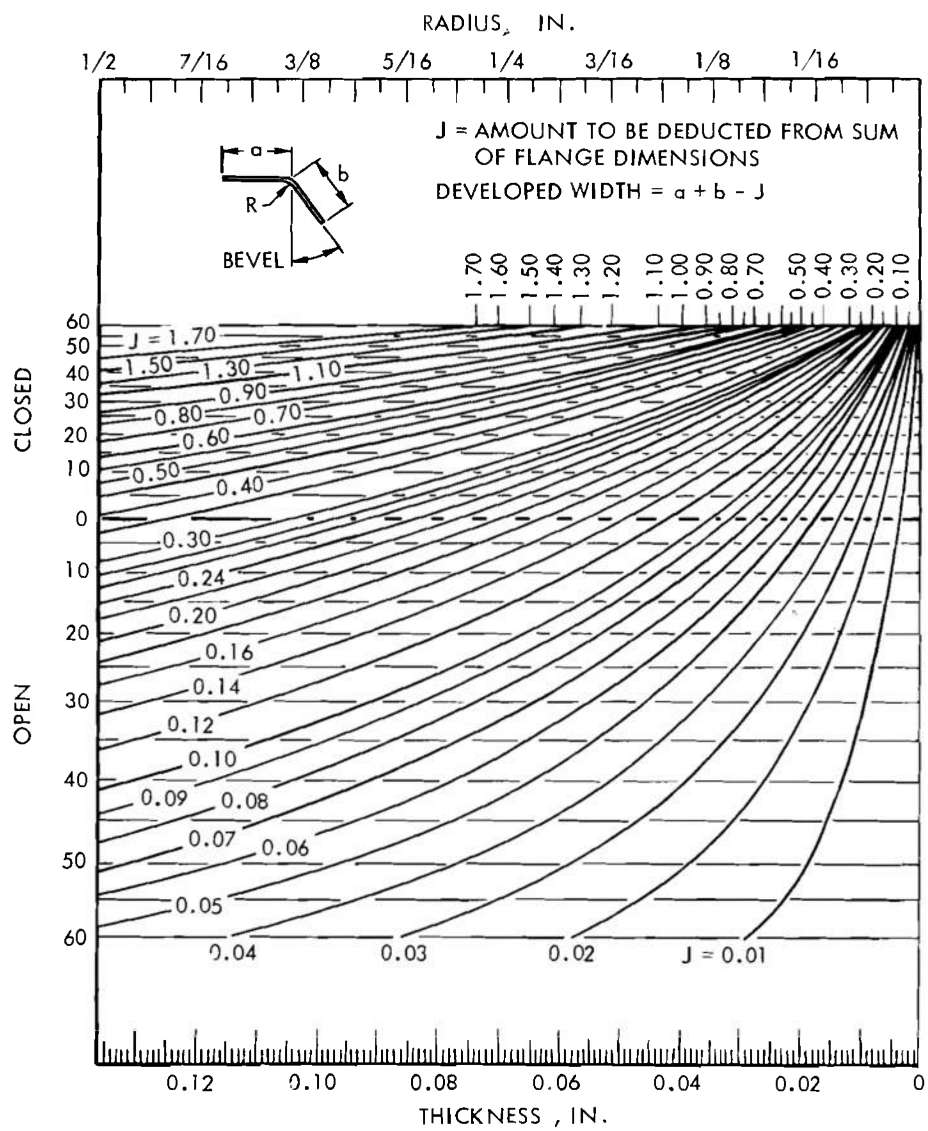


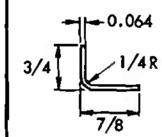
Figure 7-29. Developed Length Chart

BEND RADIUS R	MATERIAL THICKNESS T 0.016	0.020	0.025	0.032	0.040	0.051	0.064	0.072	0.081	0.091	0.125	0.156	0.188	BEND RADIUS R
1/32	0.0342	0.0395	0.0459	0.0554										1/32
1/16	0.0481	0.0525	0.0589	0.0675	0.0790	0.0929	0.1099	0.1205						1/16
3/32	0.0611	0.0664	0.0728	0.0823	0.0920	0.1068	0.1238	0.1335	0.1461	0.1589				3/32
1/8	0.0750	0.0794	0.0858	0.0953	0.1059	0.1198	0.1368	0.1474	0.1591	0.1719				1/8
5/32	0.0880	0.0933	0.0997	0.1092	0.1189	0.1337	0.1507	0.1613	0.1730	0.1858	0.2295			5/32
3/16	0.1019	0.1072	0.1127	0.1222	0.1328	0.1467	0.1637	0.1743	0.1860	0.1988	0.2434	0.2829	0.3253	3/16
7/32	0.1149	0.1202	0.1266	0.1361	0.1467	0.1606	0.1776	0.1882	0.1999	0.2127	0.2564	0.2968	0.3383	7/32
1/4	0.1288	0.1341	0.1396	0.1491	0.1597	0.1736	0.1906	0.2012	0.2129	0.2257	0.2703	0.3098	0.3522	1/4
9/32	0.1418	0.1471	0.1535	0.1630	0.1736	0.1875	0.2045	0.2151	0.2268	0.2396	0.2833	0.3237	0.3661	9/32
5/16	0.1557	0.1610	0.1674	0.1760	0.1866	0.2014	0.2175	0.2281	0.2398	0.2526	0.2972	0.3367	0.3791	5/16
11/32	0.1687	0.1740	0.1804	0.1899	0.2005	0.2144	0.2314	0.2420	0.2537	0.2665	0.3102	0.3506	0.3930	11/32
3/8	0.1826	0.1879	0.1943	0.2029	0.2135	0.2283	0.2444	0.2550	0.2667	0.2795	0.3241	0.3636	0.4060	3/8
13/32	0.1956	0.2009	0.2073	0.2168	0.2274	0.2413	0.2583	0.2689	0.2806	0.2934	0.3371	0.3775	0.4199	13/32
7/16	0.2095	0.2148	0.2212	0.2307	0.2404	0.2552	0.2722	0.2819	0.2936	0.3064	0.3510	0.3914	0.4329	7/16
15/32	0.2234	0.2278	0.2342	0.2437	0.2543	0.2682	0.2852	0.2958	0.3075	0.3203	0.3640	0.4044	0.4468	15/32
1/2	0.2364	0.2417	0.2481	0.2576	0.2673	0.2821	0.2991	0.3088	0.3205	0.3333	0.3779	0.4183	0.4598	1/2
17/32		0.2547	0.2611	0.2706	0.2812	0.2951	0.3121	0.3227	0.3344	0.3472	0.3918	0.4313	0.4737	17/32
9/16			0.2750	0.2845	0.2942	0.3090	0.3260	0.3357	0.3483	0.3611	0.4048	0.4452	0.4867	9/16
19/32				0.2993	0.3090	0.3238	0.3408	0.3496	0.3613	0.3750	0.4187	0.4582	0.5006	19/32
5/8					0.3211	0.3359	0.3529	0.3626	0.3752	0.3880	0.4317	0.4721	0.5136	5/8
21/32						0.3498	0.3668	0.3765	0.3882	0.4019	0.4456	0.4851	0.5275	21/32
11/16							0.3798	0.3904	0.4021	0.4149	0.4586	0.4990	0.5405	11/16
23/32								0.4034	0.4151	0.4297	0.4725	0.5129	0.5544	23/32
3/4									0.4290	0.4418	0.4855	0.5259	0.5674	3/4
25/32										0.4557	0.4994	0.5389	0.5813	25/32
13/16											0.5124	0.5528	0.5952	13/16
27/32											0.5263	0.5658	0.6082	27/32
7/8											0.5393	0.5797	0.6221	7/8
29/32											0.5332	0.5936	0.6351	29/32
15/16												0.6066	0.6490	15/16
31/32												0.6205	0.6620	31/32
1												0.6335	0.6759	1

NOTE:
1. ALL DIMENSIONS IN IN.
2. BEND ALLOWANCES FOR RADII GREATER THAN 1 IN. SEE FIG. 7-29

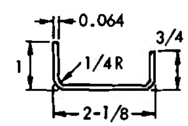
TO FIND DEVELOPED LENGTH MULTIPLY THE NUMBER OF BENDS BY THE APPLICABLE CHART VALUE AND SUBTRACT THE RESULT FROM THE SUM OF THE FLANGE DIMENSIONS.

EXAMPLE



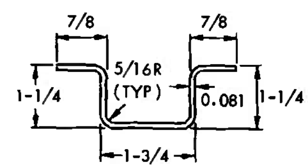
3/4 + 7/8 = 1-5/8

1.625 - 0.1906 = 1.434 DEVELOPED LENGTH



1 + 2-1/8 + 3/4 = 3-7/8

3.875 - 2(0.1906) = 3.4938 DEVELOPED LENGTH



7/8 + 1-1/4 + 1-3/4 + 1-1/4 + 7/8 = 6.000

6.000 - 4(0.2398) = DEVELOPED LENGTH

6.000 - 0.9592 = 5.041

Figure 7-30. Developed Length of 90° Bends

	MATERIAL THICKNESS T																			
BEND RADIUS R	0.016	0.019-0.020	0.025	0.028	0.031-0.032	0.038	0.040	0.050-0.051	0.063-0.064	0.072	0.078	0.081	0.091	0.094	0.109	0.125	0.156	0.188	0.250	BEND RADIUS R
1/32	0.00067	0.00070	0.00074	0.00077	0.00079	0.00084														1/32
1/16	0.00121	0.00125	0.00129	0.00131	0.00135	0.00138	0.00140	0.00149	0.00159	0.00165										1/16
3/32	0.00176	0.00179	0.00183	0.00186	0.00188	0.00193	0.00195	0.00203	0.00213	0.00220	0.00224	0.00226	0.00234							3/32
1/8	0.00230	0.00234	0.00238	0.00240	0.00243	0.00247	0.00249	0.00258	0.00268	0.00274	0.00279	0.00281	0.00289	0.00291						1/8
5/32	0.00285	0.00288	0.00292	0.00295	0.00297	0.00302	0.00304	0.00312	0.00322	0.00328	0.00333	0.00335	0.00343	0.00346	0.00358	0.00370				5/32
3/16	0.00339	0.00342	0.00347	0.00349	0.00352	0.00356	0.00358	0.00367	0.00377	0.00383	0.00388	0.00390	0.00398	0.00400	0.00412	0.00424	0.00449	0.00473		3/16
7/32	0.00394	0.00397	0.00401	0.00403	0.00406	0.00411	0.00412	0.00421	0.00431	0.00437	0.00442	0.00444	0.00452	0.00454	0.00467	0.00479	0.00503	0.00528		7/32
1/4	0.00448	0.00451	0.00456	0.00458	0.00461	0.00465	0.00467	0.00476	0.00486	0.00492	0.00497	0.00499	0.00507	0.00509	0.00521	0.00533	0.00558	0.00582		1/4
9/32	0.00503	0.00506	0.00510	0.00512	0.00515	0.00519	0.00521	0.00530	0.00540	0.00546	0.00551	0.00553	0.00561	0.00563	0.00576	0.00588	0.00612	0.00636		9/32
5/16	0.00557	0.00561	0.00564	0.00567	0.00570	0.00574	0.00576	0.00584	0.00595	0.00601	0.00606	0.00608	0.00616	0.00618	0.00630	0.00642	0.00667	0.00691		5/16
11/32	0.00612	0.00615	0.00619	0.00621	0.00624	0.00628	0.00630	0.00639	0.00649	0.00655	0.00660	0.00662	0.00670	0.00672	0.00684	0.00697	0.00721	0.00745		11/32
3/8	0.00666	0.00669	0.00673	0.00676	0.00679	0.00683	0.00685	0.00693	0.00704	0.00710	0.00715	0.00717	0.00725	0.00727	0.00739	0.00751	0.00776	0.00800	0.00849	3/8
13/32	0.00721	0.00724	0.00728	0.00730	0.00733	0.00737	0.00739	0.00748	0.00758	0.00764	0.00769	0.00771	0.00779	0.00781	0.00793	0.00806	0.00830	0.00854	0.00903	13/32
7/16	0.00775	0.00778	0.00782	0.00785	0.00787	0.00792	0.00794	0.00802	0.00812	0.00819	0.00823	0.00826	0.00834	0.00836	0.00848	0.00860	0.00884	0.00909	0.00958	7/16
15/32	0.00829	0.00833	0.00837	0.00839	0.00842	0.00846	0.00848	0.00857	0.00867	0.00873	0.00878	0.00880	0.00888	0.00890	0.00902	0.00915	0.00939	0.00963	0.01012	15/32
1/2	0.00884	0.00887	0.00891	0.00894	0.00896	0.00901	0.00903	0.00911	0.00921	0.00928	0.00932	0.00935	0.00943	0.00945	0.00957	0.00969	0.00993	0.01018	0.01067	1/2
17/32	0.00938	0.00942	0.00946	0.00948	0.00951	0.00955	0.00957	0.00966	0.00976	0.00982	0.00987	0.00989	0.00997	0.00999	0.01011	0.01023	0.01048	0.01072	0.01121	17/32
9/16	0.00993	0.00996	0.01000	0.01002	0.01005	0.01010	0.01012	0.01020	0.01030	0.01037	0.01041	0.01043	0.01051	0.01054	0.01066	0.01078	0.01102	0.01127	0.01175	9/16
19/32	0.01047	0.01051	0.01055	0.01057	0.01058	0.01064	0.01065	0.01073	0.01083	0.01091	0.01096	0.01098	0.01105	0.01108	0.01120	0.01132	0.01157	0.01181	0.01230	19/32
5/8	0.01102	0.01105	0.01109	0.01112	0.01114	0.01119	0.01121	0.01129	0.01139	0.01146	0.01150	0.01152	0.01160	0.01163	0.01175	0.01187	0.01211	0.01236	0.01284	5/8
21/32	0.01156	0.01160	0.01164	0.01166	0.01170	0.01173	0.01175	0.01183	0.01193	0.01200	0.01205	0.01207	0.01214	0.01217	0.01229	0.01241	0.01266	0.01290	0.01339	21/32
11/16	0.01211	0.01214	0.01218	0.01220	0.01223	0.01228	0.01230	0.01238	0.01248	0.01254	0.01259	0.01261	0.01269	0.01271	0.01284	0.01296	0.01320	0.01345	0.01393	11/16
23/32	0.01265	0.01268	0.01273	0.01275	0.01276	0.01282	0.01283	0.01291	0.01301	0.01309	0.01314	0.01316	0.01322	0.01326	0.01338	0.01350	0.01374	0.01399	0.01448	23/32
3/4	0.01320	0.01323	0.01327	0.01329	0.01332	0.01337	0.01338	0.01347	0.01357	0.01363	0.01368	0.01370	0.01378	0.01380	0.01393	0.01405	0.01429	0.01454	0.01502	3/4
25/32	0.01374	0.01378	0.01381	0.01384	0.01386	0.01391	0.01392	0.01401	0.01411	0.01418	0.01423	0.01425	0.01432	0.01435	0.01448	0.01459	0.01484	0.01508	0.01557	25/32
13/16	0.01429	0.01432	0.01436	0.01438	0.01441	0.01445	0.01447	0.01456	0.01466	0.01472	0.01477	0.01479	0.01487	0.01489	0.01502	0.01514	0.01538	0.01562	0.01611	13/16
27/32	0.01483	0.01486	0.01490	0.01493	0.01494	0.01500	0.01501	0.01509	0.01519	0.01527	0.01532	0.01534	0.01540	0.01544	0.01556	0.01568	0.01593	0.01617	0.01666	27/32
7/8	0.01538	0.01541	0.01545	0.01547	0.01550	0.01554	0.01556	0.01565	0.01575	0.01581	0.01586	0.01588	0.01596	0.01598	0.01610	0.01623	0.01647	0.01671	0.01720	7/8
29/32	0.01592	0.01595	0.01599	0.01602	0.01604	0.01609	0.01611	0.01619	0.01629	0.01636	0.01641	0.01643	0.01650	0.01653	0.01665	0.01677	0.01701	0.01726	0.01775	29/32
15/16	0.01646	0.01650	0.01654	0.01656	0.01659	0.01663	0.01665	0.01674	0.01684	0.01690	0.01695	0.01697	0.01705	0.01707	0.01719	0.01732	0.01756	0.01780	0.01829	15/16
31/32	0.01701	0.01704	0.01708	0.01711	0.01712	0.01718	0.01720	0.01727	0.01737	0.01745	0.01749	0.01752	0.01758	0.01762	0.01774	0.01786	0.01810	0.01835	0.01884	31/32
1	0.01755	0.01759	0.01763	0.01765	0.01768	0.01772	0.01774	0.01783	0.01793	0.01799	0.01804	0.01806	0.01814	0.01816	0.01828	0.01841	0.01865	0.01889	0.01938	1

NOTE:
1. ALL DIMENSIONS IN IN.
2. BEND ALLOWANCES FOR RADII GREATER THAN 1 IN. SEE FIG. 7-29

VALUES IN THIS TABLE ARE DERIVED FROM THE EMPIRICAL FORMULA:

$(0.01743R + 0.0078T) \times$ NUMBER OF DEGREES IN THE BEND = BEND ALLOWANCE

WHERE R = BEND RADIUS
T = MATERIAL THICKNESS

MULTIPLY VALUES SHOWN IN THIS TABLE BY THE NUMBER OF DEGREES IN A BEND TO OBTAIN BEND ALLOWANCE

Figure 7-31. Bend Allowance Chart—1° Bend

	MATERIAL THICKNESS T																			
BEND RADIUS R	0.016	0.019-0.020	0.025	0.028	0.031-0.032	0.038	0.040	0.050-0.051	0.063-0.064	0.072	0.078	0.081	0.091	0.094	0.109	0.125	0.156	0.188	0.250	BEND RADIUS R
1/32	0.0603	0.0630	0.0666	0.0693	0.0711	0.0756														1/32
1/16	0.1089	0.1125	0.1161	0.1179	0.1215	0.1242	0.1260	0.1341	0.1431	0.1485										1/16
3/32	0.1584	0.1611	0.1647	0.1674	0.1692	0.1737	0.1755	0.1827	0.1917	0.1980	0.2016	0.2034	0.2106							3/32
1/8	0.2070	0.2106	0.2142	0.2160	0.2187	0.2223	0.2241	0.2322	0.2412	0.2466	0.2511	0.2529	0.2601	0.2619						1/8
5/32	0.2565	0.2592	0.2628	0.2655	0.2673	0.2718	0.2736	0.2808	0.2898	0.2952	0.2997	0.3015	0.3087	0.3114	0.3222	0.3330				5/32
3/16	0.3051	0.3078	0.3123	0.3141	0.3168	0.3204	0.3222	0.3303	0.3393	0.3447	0.3492	0.3510	0.3582	0.3600	0.3708	0.3816	0.4041	0.4257		3/16
7/32	0.3546	0.3573	0.3609	0.3627	0.3654	0.3699	0.3708	0.3789	0.3879	0.3933	0.3978	0.3996	0.4068	0.4086	0.4203	0.4311	0.4527	0.4752		7/32
1/4	0.4032	0.4059	0.4104	0.4122	0.4149	0.4185	0.4203	0.4284	0.4374	0.4428	0.4473	0.4491	0.4563	0.4581	0.4689	0.4797	0.5022	0.5238		1/4
9/32	0.4527	0.4554	0.4590	0.4608	0.4635	0.4671	0.4689	0.4770	0.4860	0.4914	0.4959	0.4977	0.5049	0.5067	0.5184	0.5292	0.5508	0.5724		9/32
5/16	0.5013	0.5040	0.5076	0.5103	0.5130	0.5166	0.5184	0.5256	0.5355	0.5409	0.5454	0.5472	0.5544	0.5562	0.5670	0.5778	0.6003	0.6219		5/16
11/32	0.5508	0.5535	0.5571	0.5589	0.5616	0.5652	0.5670	0.5751	0.5841	0.5895	0.5940	0.5958	0.6030	0.6048	0.6156	0.6273	0.6489	0.6705		11/32
3/8	0.5994	0.6021	0.6057	0.6084	0.6111	0.6147	0.6165	0.6237	0.6336	0.6390	0.6435	0.6453	0.6525	0.6543	0.6651	0.6759	0.6984	0.7200	0.7641	3/8
13/32	0.6489	0.6516	0.6552	0.6570	0.6597	0.6633	0.6651	0.6732	0.6822	0.6876	0.6921	0.6939	0.7011	0.7029	0.7137	0.7254	0.7470	0.7686	0.8127	13/32
7/16	0.6975	0.7002	0.7038	0.7065	0.7083	0.7128	0.7146	0.7218	0.7308	0.7371	0.7407	0.7434	0.7506	0.7524	0.7632	0.7740	0.7956	0.8181	0.8622	7/16
15/32	0.7461	0.7497	0.7533	0.7551	0.7578	0.7614	0.7632	0.7713	0.7803	0.7857	0.7902	0.7920	0.7992	0.8010	0.8118	0.8235	0.8451	0.8667	0.9108	15/32
1/2	0.7956	0.7983	0.8019	0.8046	0.8064	0.8109	0.8127	0.8100	0.8289	0.8352	0.8388	0.8415	0.8487	0.8505	0.8613	0.8721	0.8937	0.9162	0.9603	1/2
17/32	0.8442	0.8478	0.8514	0.8532	0.8559	0.8595	0.8613	0.8694	0.8784	0.8838	0.8883	0.8901	0.8973	0.8991	0.9099	0.9207	0.9432	0.9648	1.0089	17/32
9/16	0.8937	0.8964	0.9000	0.9018	0.9045	0.9090	0.9108	0.9180	0.9270	0.9333	0.9369	0.9387	0.9459	0.9486	0.9594	0.9702	0.9918	1.0143	1.0575	9/16
19/32	0.9423	0.9459	0.9495	0.9513	0.9522	0.9576	0.9585	0.9657	0.9747	0.9819	0.9864	0.9882	0.9745	0.9972	1.0080	1.0188	1.0413	1.0629	1.1070	19/32
5/8	0.9918	0.9945	0.9981	1.0008	1.0026	1.0071	1.0089	1.0161	1.0251	1.0314	1.0350	1.0368	1.0440	1.0467	1.0575	1.0683	1.0899	1.1124	1.1560	5/8
21/32	1.0404	1.0440	1.0476	1.0494	1.0530	1.0557	1.0575	1.0647	1.0737	1.0800	1.0845	1.0863	1.0926	1.0953	1.1061	1.1169	1.1394	1.1610	1.2051	21/32
11/16	1.0899	1.0926	1.0962	1.0980	1.1007	1.1052	1.1070	1.1142	1.1232	1.1286	1.1331	1.1349	1.1421	1.1439	1.1556	1.1664	1.1880	1.2105	1.2537	11/16
23/32	1.1385	1.1412	1.1457	1.1475	1.1484	1.1538	1.1547	1.1619	1.1709	1.1781	1.1826	1.1844	1.1898	1.1934	1.2042	1.2150	1.2366	1.2591	1.3032	23/32
3/4	1.1880	1.1907	1.1943	1.1961	1.1988	1.2033	1.2042	1.2123	1.2213	1.2267	1.2312	1.2330	1.2402	1.2420	1.2537	1.2645	1.2861	1.3086	1.3518	3/4
25/32	1.2366	1.2402	1.2429	1.2456	1.2474	1.2519	1.2528	1.2609	1.2699	1.2762	1.2807	1.2825	1.2888	1.2915	1.3032	1.3131	1.3356	1.3572	1.4013	25/32
13/16	1.2861	1.2888	1.2924	1.2942	1.2969	1.3005	1.3023	1.3104	1.3194	1.3248	1.3293	1.3311	1.3383	1.3401	1.3518	1.3626	1.3842	1.4058	1.4499	13/16
27/32	1.3347	1.3374	1.3410	1.3437	1.3446	1.3500	1.3509	1.3581	1.3671	1.3743	1.3788	1.3806	1.3860	1.3896	1.4004	1.4112	1.4337	1.4553	1.4994	27/32
7/8	1.3842	1.3869	1.3905	1.3923	1.3950	1.3986	1.4004	1.4085	1.4175	1.4229	1.4274	1.4292	1.4364	1.4382	1.4490	1.4607	1.4823	1.5039	1.5480	7/8
29/32	1.4328	1.4355	1.4391	1.4418	1.4436	1.4481	1.4499	1.4571	1.4661	1.4724	1.4769	1.4787	1.4850	1.4877	1.4985	1.5093	1.5309	1.5534	1.5975	29/32
15/16	1.4814	1.4850	1.4886	1.4904	1.4931	1.4967	1.4985	1.5066	1.5156	1.5210	1.5255	1.5273	1.5345	1.5363	1.5471	1.5588	1.5804	1.6020	1.6461	15/16
31/32	1.5309	1.5336	1.5372	1.5399	1.5408	1.5462	1.5462	1.5543	1.5633	1.5705	1.5741	1.5768	1.5822	1.5858	1.5966	1.6074	1.6290	1.6515	1.6956	31/32
1	1.5795	1.5831	1.5867	1.5885	1.5912	1.5948	1.5966	1.6047	1.6137	1.6191	1.6236	1.6254	1.6326	1.6344	1.6452	1.6569	1.6785	1.7001	1.7442	1

NOTE:
1. ALL DIMENSIONS IN IN.
2. BEND ALLOWANCES FOR RADII GREATER THAN 1 IN. SEE FIGURE 7-29

VALUES IN THIS TABLE ARE DERIVED FROM THE EMPIRICAL FORMULA:

$(0.0173R + 0.0078T)\ 90^\circ$ = BEND ALLOWANCE

R = BEND RADIUS
T = MATERIAL THICKNESS

Figure 7-32. Bend Allowance Chart—90° Bend

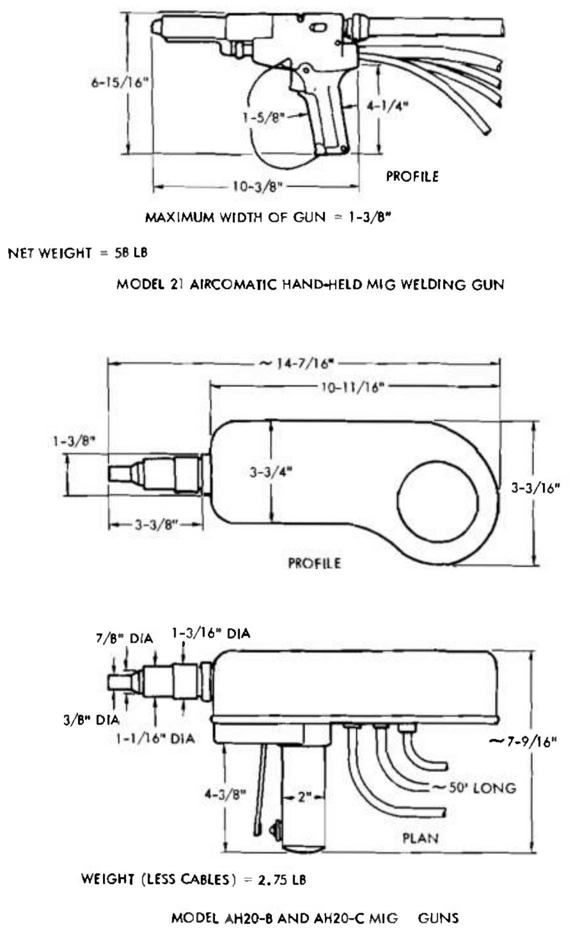


Figure 7-33. Typical MIG Hand-held Welding Guns

the stainless steel and nickel alloys. These alloys may be used for shafting, fasteners, or in the landborne running components.

Another type of reaction sometimes found on aluminum boats is exfoliation where the magnesium in the grain boundaries is attacked. This occurs primarily in the bilges when there is dirty bilge water present. Exfoliation can be controlled by special handling in the thermal part of the tempering process. This problem should be considered with the aluminum manufacturer's engineers since at this time (1969) they are still investigating this phenomenon.

Coatings for aluminum are applied using the same techniques used for steel. However, the paints used are frequently different. In particular, copper bearing antifouling paint can cause corrosion if improperly applied. An organic type of antifouling paint, such as tri-butyl-tin-oxide (TBTO), is frequently used with success and without corrosion. Aluminum amphibians are normally not left in the water for significant periods of time and therefore do not require antifouling protection. Present practice is to leave the entire vehicle unpainted both inside and out. After a period of exposure to the weather the aluminum oxide film forms on the exterior and leaves a dull gray finish which is quite satisfactory and requires no maintenance.

Table 7-12 lists the mechanical properties of the two aluminum alloys which are most frequently used for marine structures. Aluminum loses its strength without warning as it is heated. Tests show that 5052-H32 heated to 700° F for a long time will lose 90 percent of its yield strength and 47 percent at 400° F. It is important to consider in the arrangement the possible complications that could develop in a fire. For instance, the engine mounts might be safer if made from 5454 which retains its strength to a higher temperature.

Ref. 30 presents a wide variety of fatigue information which illustrates the characteristics of aluminum as a basic material and also some of the distinctive features which may appear in particular products such as extrusions, castings, and weldments. It does not, however, furnish the designers with all of the information needed for marine alloys. Fatigue tests are normally run using either axial or bending loads and with either smooth or notched specimens. The results of each variation of these tests are different so that the designer must interpret the tests for each design. The strength of a structure in a cyclic loading condition is primarily dependent on the stress level and the number of cycles. It is very important to properly estimate the number of cycles since the maximum strength in some cases will be reduced by 10,000 psi between 10^4 and 10^5 cycles. Fig. 7-34 is a modified Goodman diagram (maximum stress, minimum stress, and number of cycles) for axial stress applied to alloy 5456-0. This figure is from Ref. 34 which also contains a considerable amount of useful information on the fatigue of 5000 series aluminum. Some generalizations from Ref. 34 for 5000 series aluminum are:

a. Fatigue strength of sheet products is about 85 percent of that obtained from the rotating beam tests.

TABLE 7-16 GALVANIC SERIES

	Cathodic end of series	
	Monel	
	Nickel	
	Silicon bronze	
	Copper and copper paint	
	Aluminum bronze	
	Phosphor bronze	
Steel Construction	Manganese bronze	
Normal practical range of material	Tin	
	Lead	
	Stainless steel	Aluminum Construction
	Cast iron	Normal practical range of material
	Mild steel	
	Aluminum	
	Cadmium	
	Galvanized iron	
	Zinc	
	Magnesium	
	Anodic end of series	

b. Fatigue strength for axial-stress (complete reversal of stress) is about 80 percent of that obtained from the rotating beam tests.

c. For low numbers of cycles the fatigue strengths are generally proportional to the static strength.

d. For lives greater than 10^6 cycles, the differences in fatigue strength between cycles and tempers become less; all fall within a reasonably narrow band.

e. Most of the alloys with 4 percent or more of magnesium in the annealed temper have fatigue strengths about equal to those of cold-worked material.

f. In general, the highest fatigue properties are developed by alloy 5456.

7-3.2.5 Glass

In the past some windshields have broken from the pressures of the waves in the surf and others have cracked from unequal expansion caused by the defroster. The 3/4 in. thick laminated safety glass, approximately 23 in. wide and 13 in. high, used on the LARC V has been satisfactory. During one series of tests the glass was broken while traversing a 12 ft surf.

Fully tempered glass should be used laminated to provide non shattering characteristics. The glazing can be neoprene with a rabbet depth at least one-half of the glass thickness. The mechanical properties of tempered plate glass are listed in Table 7-12.

Electrically heated glass such as "Electropane" by Libbey-Owens-Ford Glass Company has been successfully used for de-icing and de-fogging. Applications have included jet transport planes, Coast Guard Cutters, and land vehicles. It is available in sizes up to 48 in. by 60 in.

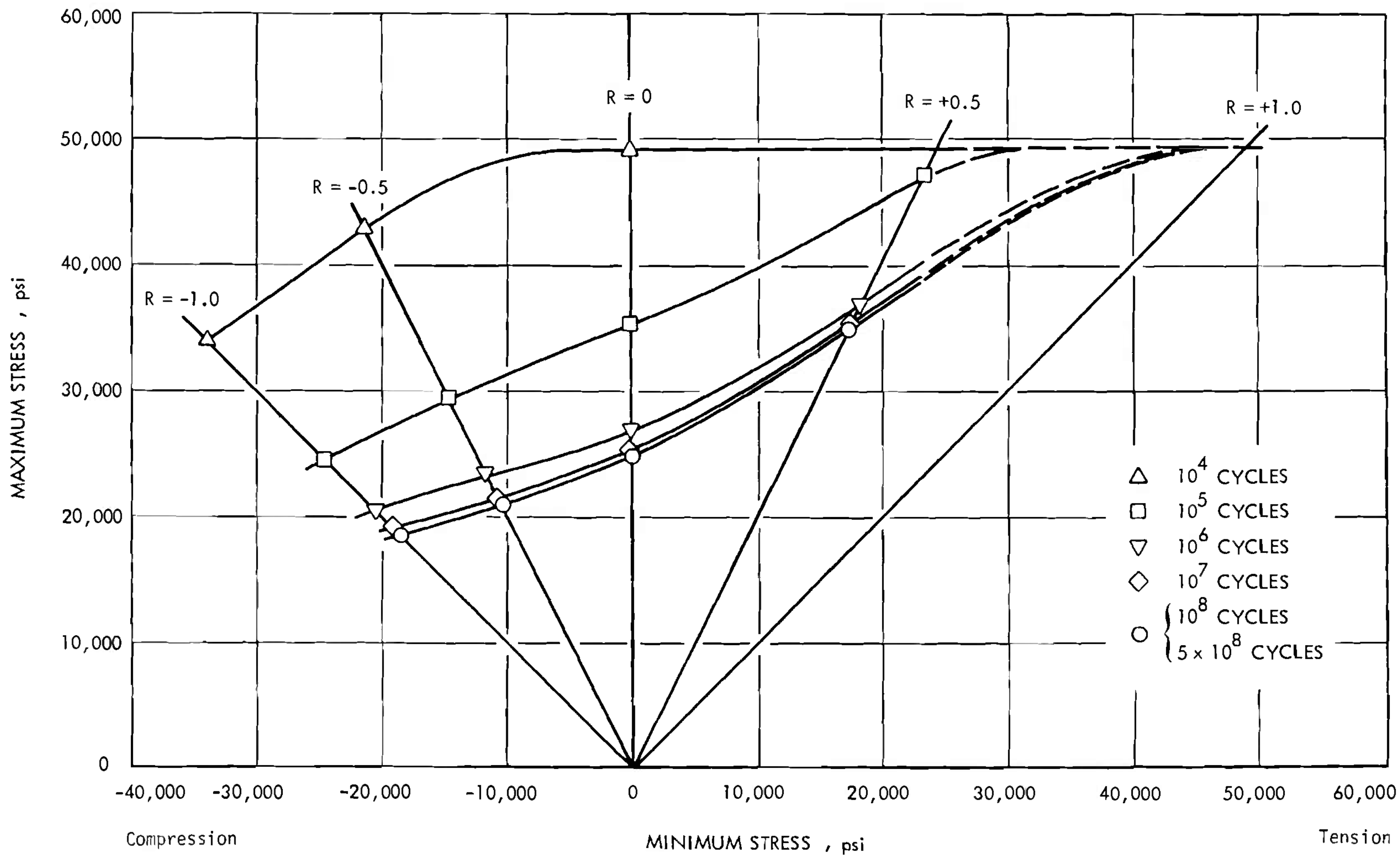


Figure 7-34. Modified Goodman Range-of-stress Diagram to Determine Typical Axial Stress Fatigue Strength for Aluminum Alloy 5456-0

7-3.3 STRUCTURAL CONFIGURATION

The primary requirements for the structure of an amphibian are twofold—

1. Support the cargo, and all of the components of engine and power transmission systems.
2. Provide a buoyant platform.

These two equally important requirements must be satisfied with respect to the total mission of the vehicle. Other considerations include:

a. Cost
b. Weight
c. Maintainability
d. Repairability
e. Waterborne resistance
f. Safety
g. Availability
h. Reliability

The starting point of the structural configuration is in preliminary design where all of the variables are considered and the best overall system selected. This is described in par. 6-5.3 in detail. It is important in the detail design to keep in mind the compromises and goals established during the preliminary design so that they are not degraded. Likewise, it is important to keep informed of the progress of the design with respect to the items that influence the structure and to inform the other members of the design team of any structural goals that cannot be met.

The most important design step is to insure compatibility of the structural configuration with the design as a whole. An expensive lightweight machinery plant with an inexpensive, heavy structure is not well engineered. An arrangement where the structure is very complex and the various parts of the hull cannot be easily joined together is also not compatible. Thus, compatibility is the first step that should be made at the start of detail design. If it is not made first, it is difficult and expensive to retrace the step later on.

The second step is a review of the access, particularly for machinery maintenance, which is a part of the maintainability problem. Access is frequently neglected in the rush to complete a design. As the design develops, access tends to get worse, rarely better. Hence if the access to any area is poor, either the arrangement must be altered or special emphasis must be applied to improve it throughout the detail structural design.

The third step is to take each load and calculate the stresses or other failure criteria in the critical structure, starting with the most severe loads and working down. It is important to consider the manner in which the forces will be transmitted through each joint. The forces should be followed through the structure until they have spread out sufficiently to become negligible. At some point in this analysis the effect of combined stresses from two or more simultaneous loads should be considered. In general, the maximum loads from a number of sources will not occur simultaneously so that it is important to apply a practical series of forces when considering simultaneous loads so as not to unduly penalize the design.

The last step in the detail structural design is to recheck the final design of those joints or intersections that are highly loaded. Those joints that have structure intersecting from 3 or more directions are particularly important because of the residual welding stresses and the 3-dimensional stresses involved.

For many designs, extra weight of structure is considered to be compatible with the design if it is used so as to reduce engineering costs and the damage from local abuse loads. This philosophy is effectively used for work boats, trucks, and other types of "low" performance structures. For this type of design there is a low cost point between the high cost of excess material and the high labor cost of complex structures. Much of the design work outlined above can be abbreviated or neglected if the structure is purposely made extra heavy.

7-3.3.1 Shell Plating

In a stressed skin configuration that uses the skin to carry the major part of the stress, the shell is the most important part of the structure. In the LARC V it is over one-third of the total structural weight. With the partial exception of kayaks and other flexible skin craft, all boats and ships use the stressed skin principle because it is the lightest configuration that will provide both the buoyancy and the structural strength required.

The forces imposed on the shell plating are:

a. Tension, compression, or shear in the plane of the shell from the hull bending

b. Shear from torsion of the hull

c. Pressure perpendicular to the shell from the water or from liquids in tanks in the hull

d. Vibration

e. Impact loads from waves and from solid objects

f. Local loads

g. Forces applied from the framing

When it is considered that all of these can act simultaneously, the list appears somewhat complex. In general, however, the maximum forces from each of these is applied to different parts of the shell. The particular combinations that are normally critical are:

a. Items a and b for shear at the mid-depth of the sides of the hull

b. Items a and c for tension or compression in the hull bottom or deck in the mid-length of the hull

c. Items a, c, and g for tension or compression in the hull bottom or deck in the mid-length where the shell is used as one flange of the frame

d. Items e and f which cover such a large variety of conditions that they must be individually considered. In general, however, their influence covers only a small area. They may be the critical load for the area and may control the design.

Refs. 35 and 36 describe the six accepted theories of failure. The four theories which apply to ductile materials state that elastic failure occurs when:

1. Maximum Stress Theory; $\sigma \geqslant \sigma_y$
2. Maximum Strain Theory; $\epsilon \geqslant \sigma_y/E$
3. Maximum Shear Theory; $\tau \geqslant \sigma_y/2$
4. Constant Energy of Distortion Theory;

$$(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2 \geqslant 2\sigma_y^2$$

where

σ = tensile or compressive stress, psi

σ_y = tensile or compressive yield stress, psi

ϵ = strain, in./in.

E = modulus of elasticity, psi

τ = shear stress, psi

$\sigma_{1,2,3}$ = principal tensile or compressive stresses, psi

Of these four theories the Constant Energy of Distortion Theory agrees best with experimental evidence but the Maximum Shear Theory leads to results which are nearly the same. The latter theory is simpler in application and is much more widely used as a basis for design. For the condition where the stress is uniaxial the Constant Energy of Distortion Theory reduces to the Maximum Stress Theory. For the design of the shell the stresses are rarely uniaxial and, therefore, the familiar Maximum Stress Theory is not adequate.

Table 7-17 lists the approximate relative allowable load for a condition where the principal stresses are all tensile and are in the ratio of 1, 2/3, and 1/3. For this particular condition, using the Maximum Stress Theory would introduce a 70 percent overload with respect to the more accurate Constant Energy of Distortion Theory. Fiberglas is an anisotropic material and must be specially considered.

Under critical loads, if the supporting structure does not fail, the shell will fail from either combined tension or compression (includes both column and bending loads), shear, or instability in compression. Shear loads are rarely critical in a stressed skin design although the torsion loads can cause trouble at the corners of large deck cut-outs such as hatches.

Consider the stresses from the hydrostatic pressure or other loads acting perpendicular to the plane of the plating. The normal boundaries of the plate form a panel which may be very small if the stiffeners are closely spaced or may cover the whole bottom of a vehicle if the framing is formed by the sides. The manner in which these panel boundaries hold the plating is a very important parameter. They may be flexible either in the plane of the panel, or perpendicular to the panel; they may be initially curved or straight; and they may hold the panel rigid at the edge or not restrict the rotation of the edge. Other major variables are the dimensions of the panel including the length L, width W, and thickness t; the type of loading including uniform, uniformly varying, or local; the characteristics of the material; and the allowable deflection. From this list it can be seen that there is a large number of possible combinations. For some, the stresses can be calculated from simple formulas; for others there are either experimental results or theory available; and for some there are no available solutions and the closest approximations to the conditions are used.

If the deflection is less than about one-half the plate thickness, the tension forces—from the

TABLE 7-17 RELATIVE ALLOWABLE LOAD AT ELASTIC FAILURE FOR FOUR THEORIES OF FAILURE

Theory of Failure	*HTS Steel*	5456-H321 *Aluminum*
Maximum Stress	1.7	1.7
Maximum Strain	1.0	0.7
Maximum Shear	2.0	1.8
Constant Energy of Distortion	1.0	1.0

deforming plate pulling in the edges—are negligible and only the stresses from bending are important. For very large deflections where the stresses exceed the yield stress, plastic flow occurs and there is a permanent set in the plate after the load is removed. In general, if a small amount of permanent set is allowed, the loads can be much higher than otherwise. For instance Ref. 37 indicates that WW II aircraft were designed to allow a permanent set of 0.005 times the shorter panel side length. This is only about one-third the plate thickness for an average span-thickness ratio, and it may be less than the welding distortion.

Table 7-18 lists the major combinations of variables and references which show their solution. Figs. 7-35 and 7-36 show how plating reacts to pressure against the plate. Solutions for three dimensional plates, i.e., curved plates, are scarce. The side on which the pressure acts is very important. If the pressure is on the concave side, the reactions are similar to those of a pressure vessel. If on the convex side, and there is some rigidity of the panel edges in the plane of the panel, the initial resistance to deflection is high but there is a critical pressure where the panel is unstable and buckling occurs. This condition can lead to "oil-canning" when the panel pops back and forth through the unstable position. From these solutions the stress at various locations in the panel that may be critical can be determined. (Fig. 7-37 is referenced in Table 7-18.)

Next consider the stresses in the panel from tension or compression of the hull acting as a beam. In tension, the stress is spread equally throughout the plate and longitudinal stiffeners. For compression, however, there is a problem of instability. For this condition the plate and stiffener combination must be considered as a unit. With transverse framing, the panels are very long and the plating can buckle between frames. The major variables for the stress in a panel in edgewise compression are:

a. Thickness/span ratio
b. Span/width ratio
c. Hydrostatic loading
d. Material characteristics
e. Plate curvature (including manufacturing distortion)
f. Edge conditions

The solution for this problem considering all of the variables is not yet available except for a few special cases. For the amphibian the normal bending stresses are low if the hull depth is reasonable in the mid-length. DDS-1100-3 and 1100-4 of Ref. 28 describe the Navy's solutions.

7-3.3.2 Stiffeners

Stressed skin construction normally implies direct and indirect framing. The direct framing is the stiffeners which support the shell, and the indirect framing is that used primarily to support the direct framing. A secondary function of the stiffeners is to support and spread the load of machinery components that are inside of the hull. If these components are small, the stresses they introduce can be neglected. It should be noted that this nomenclature is not universal and the structural components are classified differently by each industry.

The forces imposed on the stiffeners include all of those imposed on, and shared with, the shell. The types of shell-stiffener configurations are:

a. Monocoque (including sandwich)
b. Corrugated
c. Plate and stiffener (including flat bar, angle, tee, eye, channel, and inverted channel stiffeners).

7-3.3.2.1 Monocoque

Monocoque construction is a plate configuration without stiffeners. In isotropic material it is heavy except for shapes with significant curvature, light loads, or of small size. A lightweight monocoque configuration is

TABLE 7-18 FORMULAS FOR FLAT RECTANGULAR PLATES (Refs. 35 and 36)

Notation: W = total applied load, lb; M = bending moment, in.=lb; w = unit applied load, lb/in^2; t = thickness of plate, in.; s = unit stress at surface of plate, lb/in^2; y = vertical deflection of plate from original position, in.; θ = slope of plate measured from horizontal, rad; E = modulus of elasticity; m = reciprocal of ν, Poisson's ratio. q denotes any given point on the surface of plate; r denotes the distance of q from the center of a circular plate. Other dimensions and corresponding symbols are indicated on figures. Positive sign for s indicates tension at upper surface and equal compression at lower surface; negative sign indicates reverse condition. Positive sign for y indicates upward deflection, negative sign downward deflection. Subscripts r, t, a, b, f, and g used with s denote radial direction, tangential direction, direction of dimensions a, b, f, and g, respectively. All dimensions are in inches. Long side of rectangle is length (a) and the short side is length (b); aspect ratio α = b/a.

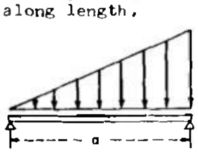
Case No.	Manner of Loading	Formulas for Stress and Deflection for Small Deflections (Max < t/2)	Formulas for use with coefficients listed at the end (ν = 0.3)
1	All edges supported. Uniform load over entire surface.	(At center) $s_a = -\dfrac{wb^2(0.225 + 0.382\alpha^2 - 0.320\alpha^3)}{t^2}$ $s_b = -\dfrac{0.75\ wb^2}{t^2(1 + 1.61\alpha^3)}$ = Max s Max $y = -\dfrac{0.1422\ wb^4}{Et^3(1 + 2.21\alpha^3)}$ (Based on coefficients given by Timoshenko, ν = 0.3)	Max $s = s_b = \dfrac{\beta wb^2}{t^2}$, at center Max $y = \dfrac{\gamma wb^4}{Et^3}$, at center
2	All edges supported. Uniform load over small concentric circular area of radius r_o.	(At center) $s_b = -\dfrac{3W}{2\pi mt^2}\left[(m+1)\ \ln\dfrac{b}{2r_o} + m(1+k)\right]$, where $k = \left(\dfrac{0.914}{1 + 1.6\alpha^5}\right) - 0.6$ Max $y = -\dfrac{0.203Wb^2(m^2 - 1)}{m^2Et^3(1 + 0.462\alpha^4)}$ (Approximate formulas based on coefficients given by Timoshenko).	Max $s = \dfrac{3W}{2\pi t^2}\left(1.3\ \ln\dfrac{2b}{\pi r_o} + 1 - \beta\right)$, at center Max $y = \dfrac{\gamma Wb^2}{Et^3}$, at center
3	All edges supported. Distributed load varying linearly along length. 	Max $s = \beta\dfrac{wb^2}{t^2}$ Max $y = \alpha\dfrac{wb^4}{Et^3}$, where β and α may be found from following table by interpolation: a/b: 1, 1.5, 2.0, 2.5, 3.0, 3.5, 4.0 β: 0.16, 0.26, 0.34, 0.38, 0.43, 0.47, 0.49 α: 0.022, 0.043, 0.060, 0.070, 0.078, 0.086, 0.091 ν = 0.3	

TABLE 7-18 (Continued)

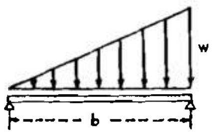
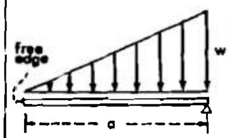
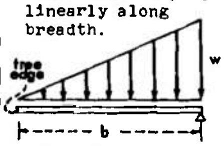
Case No.	Manner of Loading	Formulas for Stress and Deflection for Small Deflections (Max $< t/2$)	Formulas for use with coefficients listed at the end ($\nu = 0.3$)
4	All edges supported. Distributed load varying linearly along breadth. 	Max $s = \beta \frac{wb^2}{t^2}$ Max $y = \alpha \frac{wb^4}{Et^3}$, where β and α may be found from following table by interpolation: a/b: 1, 1.5, 2.0, 2.5, 3.0, 3.5, 4.0 β: 0.16, 0.26, 0.32, 0.35, 0.37, 0.38, 0.38 α: 0.022, 0.042, 0.056, 0.063, 0.067, 0.069, 0.070 $\nu = 0.3$	
5	All edges fixed. Uniform load over entire surface.	(At centers of long edges) $s_b = \frac{0.5wb^2}{t^2(1 + 0.623\alpha^6)}$ = Max s (At centers of short edges) $s_a = \frac{0.25wb^2}{t^2}$ (At center) $s_b = -\frac{0.75wb^2}{t^2(3 + 4\alpha^4)}$ $s_a = -\frac{0.054wb^2(1+2\alpha^2-\alpha^4)}{t^2}$ Max $y = -\frac{0.0284wb^4}{Et^3(1 + 1.056\alpha^5)}$ (Formula for s_b $\nu = 0.3$. Other formulas due to Westergaarde, $\nu = 0$).	Max $s = s_b = \frac{\beta wb^2}{t^2}$, at center of long edges. Max $y = \frac{\gamma wb^4}{Et^3}$, at center
6	All edges fixed. Uniform load over small concentric circular area of radius r_o.	(At center) $s_b = -\frac{3W}{2\pi m t^2}\left[(m+1)\ln\frac{b}{2r_o} + 5(1-\alpha)\right]$ = Max s, $y = \frac{\beta(m^2-1)Wb^2}{m^2Et^3}$, where β has values as follows: (Approximate stress formula by linear interpolation between stress in square plate and in infinitely long strip.	Max edge stress = $\beta W/t^2$, at center of long edges. Max $y = \gamma Wb^2/Et^3$, at center.
7	Long edges fixed, short edges supported. Uniform load over entire surface.	(At centers of long edges) $s_b = \frac{wb^2}{2t^2(1+0.2\alpha^4)}$ = Max s (At center) $s_b = -\frac{wb^2}{4t^2(1+0.4\alpha^4)}$ $s_a = -\frac{3wb^2(1+0.3\alpha^2)}{40t^2}$ ($\nu = 0$)	Max $s = s_b = \frac{\beta wb^2}{t^2}$, at center of fixed edges. Max $y = \frac{\gamma wb^4}{Et^3}$, at center

TABLE 7-18 (Continued)

<table>
<tr><th>Case No.</th><th>Manner of Loading</th><th>Formulas for Stress and Deflection for Small Deflections (Max < t/2)</th><th>Formulas for use with coefficients listed at the end ($\nu = 0.3$)</th></tr>
<tr><td>8</td><td>Short edges fixed, long edges supported. Uniform load over entire surface.</td><td>(At centers of short edges)
$s_a = \frac{0.75wb^2}{t^2(1 + 0.8\alpha^4)}$ = Max s
(At center)
$s_b = -\frac{0.75wb^2}{t^2(1 + 0.8\alpha^2 + 6\alpha^4)}$; $s_a = \frac{0.09wb^2(1 + 3\alpha^2)}{t^2(1 + \alpha^4)}$</td><td>Max $s = s_a = \frac{\beta wb^2}{t^2}$, at center of fixed edges.
Max $y = \frac{\gamma wb^4}{Et^3}$, at center</td></tr>
<tr><td>9</td><td>One long edge fixed, other free; short edges supported. Uniform load over entire surface.</td><td colspan="2">(At center of fixed edge) $s_b = \frac{3wb^2}{t^2(1 + 3.2\alpha^3)}$ = Max s
(At center of free edge) $s_a = \frac{0.75wa^2}{t^2\left(1 + \frac{0.285}{\alpha^4}\right)}$
Max $y = -\frac{1.37wb^4}{Et^3(1 + 10\alpha^3)}$
($\nu = 0.3$)</td></tr>
<tr><td>10</td><td>One long edge clamped. Other three edges supported. Uniform load over entire surface.</td><td colspan="2">Max $s = \beta \frac{wb^2}{t^2}$ Max $y = \alpha \frac{wb^4}{Et^3}$, where β and α may be found from the following table by interpolation:<table><tr><td>a/b</td><td>1</td><td>1.5</td><td>2.0</td><td>2.5</td><td>3.0</td><td>3.5</td><td>4.0</td></tr><tr><td>β</td><td>0.50</td><td>0.66</td><td>0.73</td><td>0.74</td><td>0.74</td><td>0.75</td><td>0.75</td></tr><tr><td>α</td><td>0.030</td><td>0.046</td><td>0.054</td><td>0.056</td><td>0.057</td><td>0.058</td><td>0.058</td></tr></table>($\nu = 0.3$)</td></tr>
<tr><td>11</td><td>One short edge clamped. Other three edges supported. Uniform load over entire surface.</td><td colspan="2">Max $s = \beta \frac{wb^2}{t^2}$, Max $y = \alpha \frac{wb^4}{Et^3}$, where β and α may be found from the following table by interpolation:<table><tr><td>a/b</td><td>1</td><td>1.5</td><td>2.0</td><td>2.5</td><td>3.0</td><td>3.5</td><td>4.0</td></tr><tr><td>β</td><td>0.50</td><td>0.67</td><td>0.73</td><td>0.74</td><td>0.75</td><td>0.75</td><td>0.75</td></tr><tr><td>α</td><td>0.030</td><td>0.071</td><td>0.101</td><td>0.122</td><td>0.132</td><td>0.137</td><td>0.139</td></tr></table>($\nu = 0.3$)</td></tr>
<tr><td>12</td><td>One short edge free. Other three edges supported. Uniform load over entire surface</td><td colspan="2">Max $s = \beta \frac{wb^2}{t^2}$, Max $y = \alpha \frac{wb^4}{Et^3}$, where β and α may be found from the following table by interpolation:<table><tr><td>a/b</td><td>1</td><td>1.5</td><td>2.0</td><td>4.0</td></tr><tr><td>β</td><td>0.67</td><td>0.77</td><td>0.79</td><td>0.80</td></tr><tr><td>α</td><td>0.140</td><td>0.160</td><td>0.165</td><td>0.167</td></tr></table>($\nu = 0.3$)</td></tr>
</table>

TABLE 7-18 (Continued)

Case No.	Manner of Loading	Formulas for Stress and Deflection for Small Deflections (Max < t/2)
13	One short edge free. Other three edges supported. Distributed load varying linearly along length.  	$\text{Max } s = \beta \frac{wb^2}{t^2}$, $\text{Max } y = \alpha \frac{wb^4}{Et^3}$, where β and α may be found from the following table by interpolation: a/b: 1, 1.5, 2.0, 2.5, 3.0, 3.5, 4.0 β: 0.2, 0.28, 0.32, 0.35, 0.36, 0.37, 0.37 α: 0.040, 0.050, 0.058, 0.064, 0.067, 0.069, 0.070 $(\nu = 0.3)$
14	One long edge free. Other three edges supported. Uniform load over entire surface.	$\text{Max } s = \beta \frac{wa^2}{t^2}$, $\text{Max } y = \alpha \frac{wa^4}{Et^3}$, where β and α may be found from the following table by interpolation: a/b: 1, 1.5, 2.0 β: 0.67, 0.45, 0.36 α: 0.140, 0.106, 0.080 $(\nu = 0.3)$
15	One long edge free. Other three edges supported. Distributed load, varying linearly along breadth.  	$\text{Max } s = \beta \frac{wa^2}{t^2}$, $\text{Max } y = \alpha \frac{wa^4}{Et^3}$, where β and α may be found from the following table by interpolation: a/b: 1, 1.5, 2.0 β: 0.2, 0.16, 0.11 α: 0.040, 0.033, 0.026 $(\nu = 0.3)$

Coefficients β and γ for rectangular plates with small deflections - equations above.

Case No.	Coef.	a/b = 1	1.1	1.2	1.3	1.4	1.5	1.6	1.7	1.8	1.9	2	∞
1	β	0.2874	0.3318	0.3756	0.4158	0.4518	0.4872	0.5172	0.5448	0.5688	0.5910	0.6102	0.7500
	γ	0.0443	0.0530	0.0616	0.0697	0.0770	0.0843	0.0906	0.0964	0.1017	0.1064	0.1106	0.1422
2	β	0.564	0.440	0.349	0.275	0.211	0.165	0.124	0.0950	0.072	0.0560	0.041	0.000
	γ	0.1265	0.1381	0.1478	0.1560	0.1621	0.1671	0.1714	0.1746	0.1769	0.1787	0.1803	0.1849
5	β	0.3078	0.3486	0.3834	0.4122	0.4356	0.4542	0.4680	0.4794	0.4872	0.4932	0.4974	
	γ	0.0138	0.0164	0.0188	0.0209	0.0226	0.0240	0.0251	0.0260	0.0267	0.0272	0.0277	
6	β	0.7542	0.8310	0.8940	0.9380	0.9624	0.9810	0.9906	0.9960	1.000	1.000	1.004	
	γ	0.0611	0.0666	0.0706	0.0735	0.0755	0.0768	0.0777	0.0782	0.0786	0.0787	0.0788	
7	β	0.420	0.444	0.462	0.474	0.486	0.498	0.500	0.501	0.502	0.503	0.504	0.498
	γ	0.0209	0.0228	0.0243	0.0255	0.0262	0.0270	0.0273	0.0276	0.0279	0.0282	0.0284	0.0284
8	β	0.420	0.474	0.522	0.564	0.600	0.630	0.654	0.6720	0.690	0.702	0.714	0.750
	γ	0.0209	0.0274	0.0340	0.0424	0.0502	0.0582	0.0658	0.0730	0.0799	0.0863	0.0987	0.1422

TABLE 7-18 (Continued)

Case No.	Manner of Loading	Formulas for Stress and Deflection for Small Deflections (Max < t/2)
16	Simply supported by 2 opposite edges rigid and the other 2 on elastic beams. (See Fig. 7-37) Uniform load over entire surface.	H is the ratio of the stiffness of the plate to the stiffness of the beams. $H = (1-\nu^2)\frac{E_{beam}\,I_{beam}}{E_{plate}\,I_{plate}}$

Values of C Used in Determining Bending Moment in the (f) direction at the Center from Equation

$$M_f = C\frac{wf^2}{8}\ ;\ \text{Poisson's Ratio } \nu = 0^*$$

Values of g/f	Values of C								
	H = 0	H = 0.05	H = 0.10	H = 0.20	H = 0.50	H = 1.0	H = 2.0	H = 4.0	H = ∞
0.33	1.015	0.751	0.625	0.452	0.250	0.145	0.081	0.045	0.006
0.5	1.008	0.834	0.713	0.555	0.342	0.222	0.136	0.088	0.035
0.6	1.004	0.856	0.748	0.603	0.394	0.267	0.180	0.129	0.071
0.8	0.997	0.886	0.802	0.681	0.496	0.373	0.286	0.232	0.171
1.0	0.992	0.908	0.843	0.745	0.588	0.481	0.402	0.353	0.295
1.2	0.990	0.925	0.874	0.796	0.669	0.579	0.512	0.470	0.418
1.4	0.990	0.939	0.899	0.838	0.739	0.664	0.609	0.574	0.532
1.6	0.991	0.951	0.920	0.873	0.793	0.735	0.691	0.662	0.628
1.8	0.991	0.961	0.936	0.900	0.837	0.791	0.757	0.735	0.707

Values of C Used in Determining Bending Moment in the (g) direction at the Center from Equation

$$M_g = C\frac{wf^2}{8}\ ;\ \text{Poisson's Ratio } \nu = 0^*$$

0.33	-0.125	-0.052	-0.038	0.001	0.039	0.100	0.085	0.091	0.101
0.5	-0.098	-0.045	-0.009	0.037	0.101	0.138	0.167	0.177	0.197
0.6	-0.092	-0.032	+0.005	0.054	0.126	0.169	0.198	0.217	0.236
0.8	-0.055	-0.009	+0.026	0.076	0.152	0.203	0.239	0.262	0.287
1.0	-0.035	-0.060	+0.037	0.082	0.155	0.207	0.243	0.268	0.295
1.2	-0.021	+0.012	+0.039	0.080	0.145	0.192	0.227	0.250	0.275
1.4	-0.012	+0.016	+0.038	0.073	0.129	0.170	0.199	0.219	0.242
1.6	-0.007	+0.016	+0.035	0.063	0.110	0.144	0.170	0.186	0.206
1.8	-0.003	+0.016	+0.031	0.053	0.091	0.120	0.140	0.154	0.171

* The values of the bending moment for any value of ν may be found by using Bending Moment with Poisson's Ratio Effect equal to the bending moment without Poisson's Ratio plus the product of the bending moment at the same location in a perpendicular plane times Poisson's ratio.

TABLE 7-18 (Continued)

Values of C Used in Determining Deflection at Center of Plate from Equation

$$y = C(1-\nu^2)\,\frac{wf^4}{Et^3}\;;\ \text{Poisson's Ratio}\ \nu = 0.15$$

Values of g/f	Values of C								
	H = 0	H = 0.05	H = 0.10	H = 0.20	H = 0.50	H = 1.0	H = 2.0	H = 4.0	H = ∞
0.33	0.1591	0.1214	0.0983	0.0714	0.0398	0.0234	0.0135	0.0079	0.0019
0.5	0.1579	0.1310	0.1122	0.0879	0.0549	0.0358	0.0233	0.0160	0.0079
0.6	0.1573	0.1345	0.1180	0.0956	0.0636	0.0439	0.0307	0.0228	0.0140
0.8	0.1563	0.1394	0.1264	0.1080	0.0796	0.0609	0.0476	0.0395	0.0301
1.0	0.1557	0.1428	0.1327	0.1179	0.0940	0.0776	0.0656	0.0581	0.0493
1.2	0.1554	0.1455	0.1314	0.1258	0.1064	0.0927	0.0825	0.0761	0.0684
1.4	0.1553	0.1476	0.1415	0.1322	0.1167	0.1057	0.0974	0.0921	0.0856
1.6	0.1554	0.1495	0.1446	0.1374	0.1253	0.1165	0.1098	0.1056	0.1004
1.8	0.1556	0.1509	0.1473	0.1416	0.1321	0.1252	0.1200	0.1165	0.1125
2.0	0.1557	0.1521	0.1493	0.1449	0.1375	0.1321	0.1280	0.1253	0.1222
3.0	0.1565	0.1557	0.1551	0.1543	0.1528	0.1517	0.1509	0.1503	0.1496

Values of C Used in Determining Maximum Bending Moment in Flexible Beam Support from Equation $M_{beam} = Cwf^3$; Poisson's Ratio $\nu = 0.15$

Values of g/f	Values of C								
	H = 0	H = 0.05	H = 0.10	H = 0.20	H = 0.50	H = 1.0	H = 2.0	H = 4.0	H = ∞
0.33	0	0.0033	0.0052	0.0076	0.0103	0.0117	0.0126	0.0131	0.0134
0.5	0	0.0040	0.0067	0.0103	0.0152	0.0180	0.0199	0.0210	0.0217
0.6	0	0.0055	0.0095	0.0150	0.0228	0.0276	0.0308	0.0328	0.0349
0.8	0	0.0057	0.0101	0.0164	0.0261	0.0325	0.0370	0.0398	0.0430
1.0	0	0.0058	0.0104	0.0172	0.0282	0.0357	0.0412	0.0446	0.0487
1.2	0	0.0059	0.0106	0.0177	0.0294	0.0376	0.0438	0.0477	0.0524
1.4	0	0.0059	0.0107	0.0179	0.0301	0.0388	0.0454	0.0496	0.0546
1.6	0	0.0060	0.0108	0.0181	0.0305	0.0395	0.0463	0.0507	0.0560
1.8	0	0.0060	0.0108	0.0182	0.0308	0.0399	0.0469	0.0514	0.0568
2.0	0	0.0060	0.0108	0.0182	0.0309	0.0402	0.0472	0.0518	0.0573
∞	0	0.0060	0.0109	0.0183	0.0311	0.0405	0.0477	0.0523	0.0579

Values of C Used in Determining Maximum Deflection of Flexible Beam Support from Equation $y_{beam} = C(1-\nu^2)\ wf^4/Et^3$; Poisson's Ratio $\nu = 1/4$

Values of g/f	Values of C								
	H = 0	H = 0.05	H = 0.10	H = 0.20	H = 0.50	H = 1.0	H = 2.0	H = 4.0	H = 10.0
0.33	0.146	0.1265	0.1013	0.0724	0.0390	0.0221	0.0118	0.0061	0.0049
0.5	0.170	0.1381	0.1162	0.0882	0.0512	0.0301	0.0165	0.0087	0.0070
0.6	0.172	0.1425	0.1220	0.0947	0.0567	0.0340	0.0189	0.0100	0.0081
0.8	0.174	0.1482	0.1296	0.1037	0.0648	0.0398	0.0225	0.0120	0.0098
1.0	0.175	0.1515	0.1341	0.1090	0.0698	0.0436	0.0249	0.0134	0.0109
1.2	0.175	0.1534	0.1366	0.1121	0.0728	0.0460	0.0265	0.0143	0.0117
1.4	0.176	0.1546	0.1382	0.1140	0.0747	0.0474	0.0274	0.0149	0.0121
1.6	0.176	0.1553	0.1391	0.1150	0.0757	0.0482	0.0280	0.0152	0.0124
1.8	0.176	0.1557	0.1396	0.1156	0.0763	0.0487	0.0283	0.0154	0.0125
2.0	0.176	0.1559	0.1399	0.1160	0.0767	0.0490	0.0285	0.0155	0.0126
3.0	0.189	0.1562	0.1403	0.1165	0.0772	0.0494	0.0287	0.0156	0.0127

TABLE 7-18 (Continued)

Case No.	Manner of Loading	Formulas for Stress and Deflection for Small Deflections (Max < t/2)
17	Simply supported by 2 opposite edges rigid and the other 2 on elastic beams. Load P at center uniformly distributed over a circle of diameter $d_o = f/4$.	H is the same as for case 16. Poisson's ratio correction is the same as for Case 16.

Values of C Used in Determining Bending Moment in the (f) direction at the Center from Equation $M_f = CP$; Poisson's Ratio $\nu = 0$

Values of g/f	Values of C								
	H = 0	H = 0.05	H = 0.10	H = 0.20	H = 0.50	H = 1.0	H = 2.0	H = 4.0	H = ∞
0.33	0.686	0.521	0.421	0.305	0.171	0.101	0.060	0.037	0.010
0.5	0.468	0.396	0.348	0.284	0.200	0.148	0.119	0.100	0.080
0.6	0.398	0.350	0.314	0.268	0.202	0.161	0.133	0.117	0.098
0.8	0.318	0.292	0.274	0.246	0.204	0.176	0.158	0.145	0.132
1.0	0.274	0.260	0.249	0.232	0.207	0.189	0.176	0.168	0.159
1.2	0.248	0.242	0.236	0.225	0.209	0.198	0.190	0.184	0.177
1.4	0.235	0.230	0.225	0.220	0.210	0.204	0.198	0.195	0.199
1.6	0.225	0.223	0.221	0.217	0.211	0.207	0.204	0.202	0.200
1.8	0.220	0.218	0.217	0.215	0.211	0.208	0.207	0.205	0.204
2.0	0.217	0.216	0.215	0.214	0.211	0.210	0.209	0.208	0.207
3.0	0.212	0.212	0.212	0.212	0.212	0.212	0.212	0.212	0.212

Values of C Used in Determining Bending Moment in the (g) direction at the Center from Equation $M_g = CP$; Poisson's Ratio $\nu = 0$

	Values of C								
0.33	-0.0185	0.0276	0.0553	0.0864	0.124	0.142	0.154	0.160	0.168
0.5	0.0307	0.0522	0.0687	0.0870	0.114	0.130	0.139	0.144	0.151
0.6	0.0543	0.0717	0.0840	0.101	0.124	0.138	0.147	0.163	0.160
0.8	0.0850	0.0963	0.104	0.117	0.135	0.144	0.152	0.158	0.164
1.0	0.103	0.111	0.116	0.123	0.135	0.143	0.149	0.153	0.159
1.2	0.114	0.118	0.121	0.126	0.134	0.140	0.144	0.147	0.150
1.4	0.120	0.123	0.125	0.128	0.133	0.137	0.140	0.142	0.144
1.6	0.123	0.126	0.127	0.129	0.132	0.135	0.136	0.138	0.139
1.8	0.127	0.127	0.129	0.130	0.132	0.134	0.135	0.135	0.136
2.0	0.129	0.130	0.130	0.131	0.132	0.133	0.134	0.135	0.135
3.0	0.131	0.131	0.131	0.131	0.131	0.131	0.131	0.131	0.131

If d_o is less than $f/4$, a correction of $0.17P\,(1/4 - d_o/f)/(1/4 + 1/2\, d_o/f)$ must be added to both M_f and M_g.

TABLE 7-18 (Continued)

Values of C Used in Determining Maximum Deflection at Center of Plate from Equation

$$y = C(1 - v^2) \frac{Pf^2}{Et^3} \; ; \text{ Poisson's ratio } v = 0.15$$

	Values of C								
0.33	0.7593	0.5810	0.4718	0.3449	0.1957	0.1187	0.0718	0.0456	0.0174
0.5	0.5076	0.4250	0.3677	0.2932	0.1925	0.1340	0.0958	0.0736	0.0489
0.6	0.4264	0.3693	0.3280	0.2720	0.1910	0.1429	0.1098	0.0902	0.0689
0.8	0.3305	0.3004	0.2774	0.2446	0.1942	0.1610	0.1374	0.1230	0.1063
1.0	0.2789	0.2619	0.2485	0.2290	0.1975	0.1758	0.1600	0.1501	0.1385
1.2	0.2491	0.2391	0.2312	0.2195	0.2000	0.1863	0.1761	0.1696	0.1619
1.4	0.2312	0.2253	0.2206	0.2135	0.2016	0.1930	0.1866	0.1826	0.1776
1.6	0.2202	0.2167	0.2139	0.2096	0.2024	0.1972	0.1932	0.1907	0.1877
1.8	0.2135	0.2114	0.2097	0.2071	0.2028	0.1996	0.1972	0.1957	0.1938
2.0	0.2093	0.2081	0.2071	0.2055	0.2029	0.2010	0.1996	0.1887	0.1975
3.0	0.2031	0.2030	0.2030	0.2029	0.2028	0.2027	0.2027	0.2026	0.2026

Values of C Used in Determining Maximum Bending Moment in Flexible Beam Support from Equation $M_{beam} = C\ (Pf/4)$; Poisson's Ratio $v = 0.15$

Values of g/f	Values of C								
	H = 0	H = 0.05	H = 0.10	H = 0.20	H = 0.50	H = 1.0	H = 2.0	H = 4.0	H = ∞
0.33	0	0.105	0.169	0.242	0.328	0.372	0.399	0.413	0.423
0.5	0	0.072	0.122	0.185	0.271	0.320	0.352	0.371	0.383
0.6	0	0.060	0.103	0.161	0.242	0.292	0.325	0.345	0.368
0.8	0	0.043	0.076	0.122	0.193	0.240	0.273	0.293	0.316
1.0	0	0.032	0.057	0.094	0.153	0.193	0.223	0.241	0.263
1.2	0	0.024	0.044	0.073	0.120	0.154	0.179	0.195	0.213
1.4	0	0.019	0.033	0.057	0.094	0.121	0.142	0.155	0.170
1.6	0	0.015	0.026	0.044	0.074	0.095	0.111	0.121	0.134
1.8	0	0.011	0.020	0.034	0.057	0.074	0.087	0.095	0.104

Values of C Used in Determining Maximum Deflection of Flexible Beam Support from Equation $y_{beam} = C(1-v^2) \frac{Pf^2}{Et^3}$; Poisson's Ratio $v = 0.15$

	Values of C								
0.33	0.7641	0.5805	0.4681	0.3375	0.1837	0.1044	0.0558	0.0291	0
0.5	0.5008	0.4104	0.3477	0.2664	0.1565	0.0928	0.0511	0.0269	0
0.6	0.4110	0.3454	0.2979	0.2336	0.1412	0.0857	0.0479	0.0254	0
0.8	0.2937	0.2541	0.2240	0.1810	0.1150	0.0715	0.0407	0.0219	0
1.0	0.2189	0.1923	0.1715	0.1409	0.0919	0.0582	0.0335	0.0182	0
1.2	0.1665	0.1475	0.1324	0.1098	0.0727	0.0465	0.0271	0.0150	0
1.4	0.1278	0.1137	0.1024	0.0855	0.0571	0.0368	0.0215	0.0117	0
1.6	0.0985	0.0879	0.0793	0.0664	0.0446	0.0288	0.0169	0.0092	0
1.8	0.0760	0.0679	0.0614	0.0515	0.0347	0.0235	0.0142	0.0082	0
2.0	0.0596	0.0534	0.0484	0.0408	0.0279	0.0194	0.0112	0.0066	0
3.0	0.0156	0.0140	0.0126	0.0106	0.0072	0.0047	0.0027	0.0015	0

TABLE 7-18 (Concluded)

For large deflections ($y \geq t/2$) of rectangular plates under uniform load use the following table: where, s_d = diaphragm stress, s_b = bending stress, s = total stress ($s = s_d + s_b$), and $\nu = 0.316$. All other terms are as defined at the head of this table.

a/b	Edges and point of Max s		Wb^4/Et^4										
		Coef.	0	12.5	25	50	75	100	125	150	175	200	250
1	Held; not fixed	y/t	0	0.430	0.650	0.930	1.13	1.26	1.37	1.47	1.56	1.63	1.77
	At center of plate	$s_d b^2/Et^2$	0	0.70	1.60	3.00	4.00	5.00	6.10	7.00	7.95	8.60	10.20
		sb^2/Et^2	0	3.80	5.80	8.70	10.90	12.80	14.30	15.60	17.00	18.20	20.50
1	Held; riveted	y/t	0	0.406	0.600	0.840	1.00	1.13	1.23	1.31	1.40	1.46	1.58
	At center of plate	$s_d b^2/Et^2$	0	0.609	1.380	2.68	3.80	4.78	5.75	6.54	7.55	8.10	9.53
		sb^2/Et^2	0	3.19	5.18	7.77	9.72	11.34	12.80	14.10	15.40	16.40	18.40
1	Held and fixed	y/t	0	0.165	0.25	0.59	0.80	0.95	1.08	1.19	1.28	1.38	1.54
	At center of long edges	$s_d b^2/Et^2$	0	0.070	0.22	0.75	1.35	2.00	2.70	3.30	4.00	4.60	5.90
		sb^2/Et^2	0	3.80	6.90	14.70	21.0	26.50	31.50	36.20	40.70	45.00	53.50
	At center of plate	$s_d b^2/Et^2$	0	0.075	0.30	0.95	1.65	2.40	3.10	3.80	4.50	5.20	6.50
		sb^2/Et^2	0	1.80	3.50	6.60	9.20	11.60	13.0	14.50	15.80	17.10	19.40
1.5	Held; not fixed	y/t	0	0.625	0.879	1.18	1.37	1.53	1.68	1.77	1.88	1.96	2.12
	At center of plate	$s_d b^2/Et^2$	0	1.06	2.11	3.78	5.18	6.41	7.65	8.60	9.55	10.60	12.30
		sb^2/Et^2	0	4.48	6.81	9.92	12.25	14.22	16.0	17.50	18.90	20.30	22.80
2 to ∞	Held; not fixed	y/t	0	0.696	0.946	1.24	1.44	1.60	1.72	1.84	1.94	2.03	2.20
	At center of plate	$s_d b^2/Et^2$	0	1.29	2.40	4.15	5.61	6.91	8.10	9.21	10.10	10.90	13.20
		sb^2/Et^2	0	4.87	7.16	10.30	12.60	14.60	16.40	18.00	19.40	20.90	23.60
1.5 to ∞	Held and fixed	y/t	0	0.28	0.51	0.825	1.07	1.24	1.40	1.50	1.63	1.72	1.86
	At center of long edges	$s_d b^2/Et^2$	0	0.20	0.66	1.90	3.20	4.35	5.40	6.50	7.50	8.50	10.30
		sb^2/Et^2	0	5.75	11.12	20.30	27.8	35.0	41.0	47.0	52.50	57.60	67.00

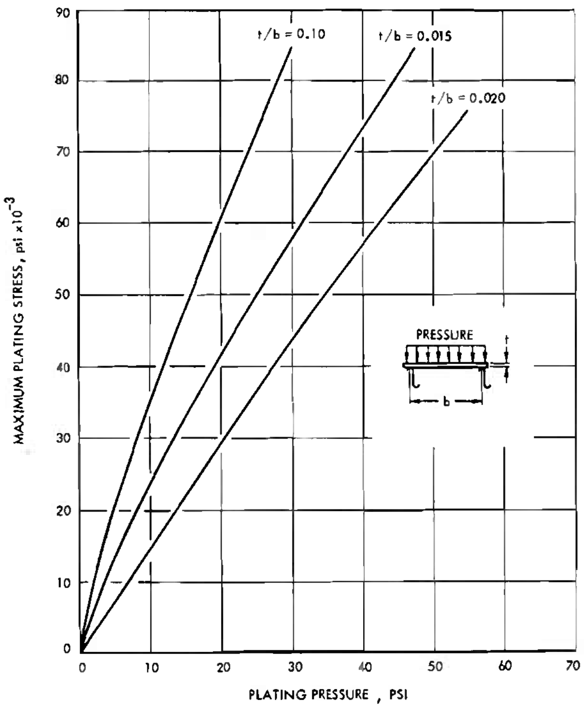


Figure 7-35. Plating Stress as a Function of Plating Pressure

sandwich construction where the inner and outer faces carry the tension and compression, and the lightweight core carries only the shear load. Sandwich-type construction is used with Fiberglas or metal for the face plates. The purpose of this arrangement is to increase the rigidity and strength of a thin plate with a minimum increase in weight. The problems anticipated for use in an amphibian include:

a. Leaks into the core

b. Difficulty of fastening to the faces and edges

c. Ability to absorb shear between the core and the faces

All of these can be solved and sandwich construction is used for boats and missiles. Core materials used are end-grain balsa-wood, paper or metal honeycomb, and a variety of types of plastic foam. Many of the larger Fiberglas work boats, such as trawlers, are now being built with a sandwich configuration.

The theoretical analysis of sandwich construction is a lengthy process and the theory is not yet complete. The analysis generally results in expressions which are very cumbersome and tedious to evaluate. Ref. 29 presents a few simplified examples of the design of sandwich panels. This type of construction is undergoing a high rate of development and the latest references should be consulted for design information.

7-3.3.2.2 Corrugations

Corrugations add stiffness and eliminate the need for stiffeners. They also leave a hydrodynamically rough surface and therefore, are rarely used for the exterior of boat or ship hulls. They may be used for bulkheads and other interior components and, for amphibians, they could be considered for use on the sides, deck, or interior structure. In general, the weight of corrugated structure is higher than the minimum possible with other configurations. It is also less expensive than conventional structural systems.

The simple methods of analysis which follow (Ref. 38) are sufficient to estimate the scantlings for a corrugated structure. For areas with high loadings, additional studies should be made to determine that the structural stability is sufficient for the use intended.

The equations which follow are for pressed corrugations. Sheets corrugated in this manner do not undergo any reduction in width. Horizontal parts remain as thick as they were in the original sheet, but the webs become thinner as they are formed by stretching. Another method of corrugation reduces the width of the finished sheet but retains a constant thickness throughout. These are formed by bending or rolling. The equations hold for these constant-thickness corrugations, except, that as the corrugations become nearly rectangular, the inertia of the web is underestimated, hence, the equations are conservative.

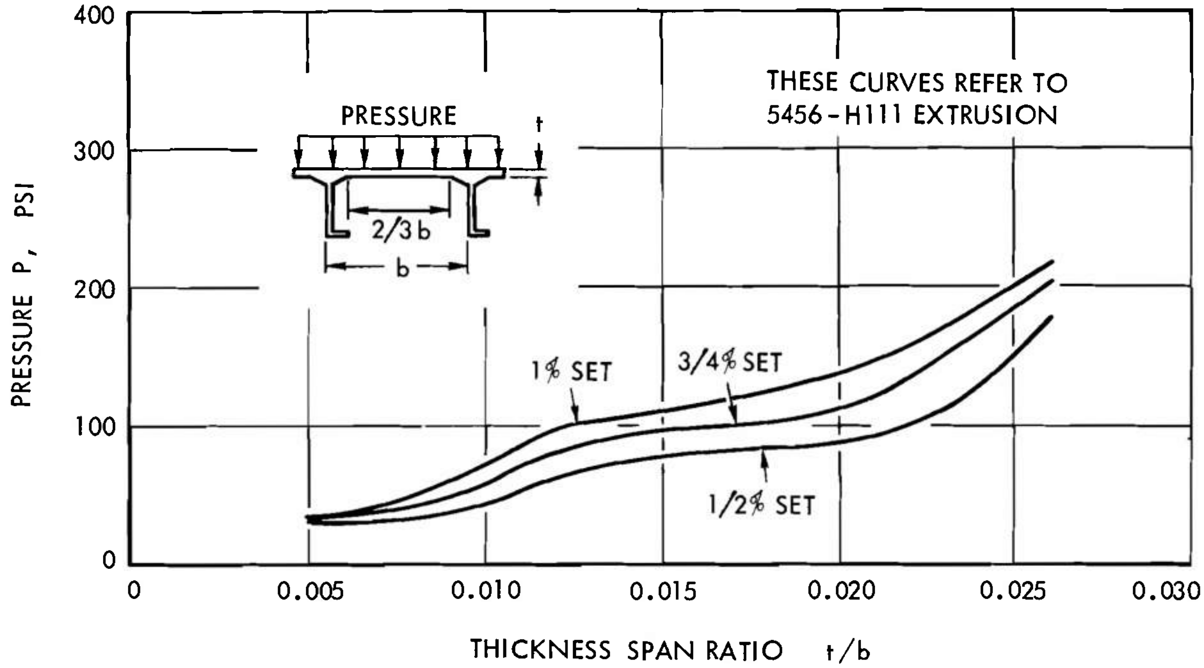
Height of neutral axis above reference line ($y = 0$)

$$\bar{y} = K_1 t\,(2 - K_3 - K_2 K_3)/2,\ \text{in.} \qquad (7\text{-}41)$$

Moment of inertia

$$I = Lt^3\,(K_1^2 A + 1)/12,\ \text{in}^4 \qquad (7\text{-}42)$$

where the symbols are defined in Fig. 7-38. When webs are curved, a straight line approximation is used for the webs and flanges.



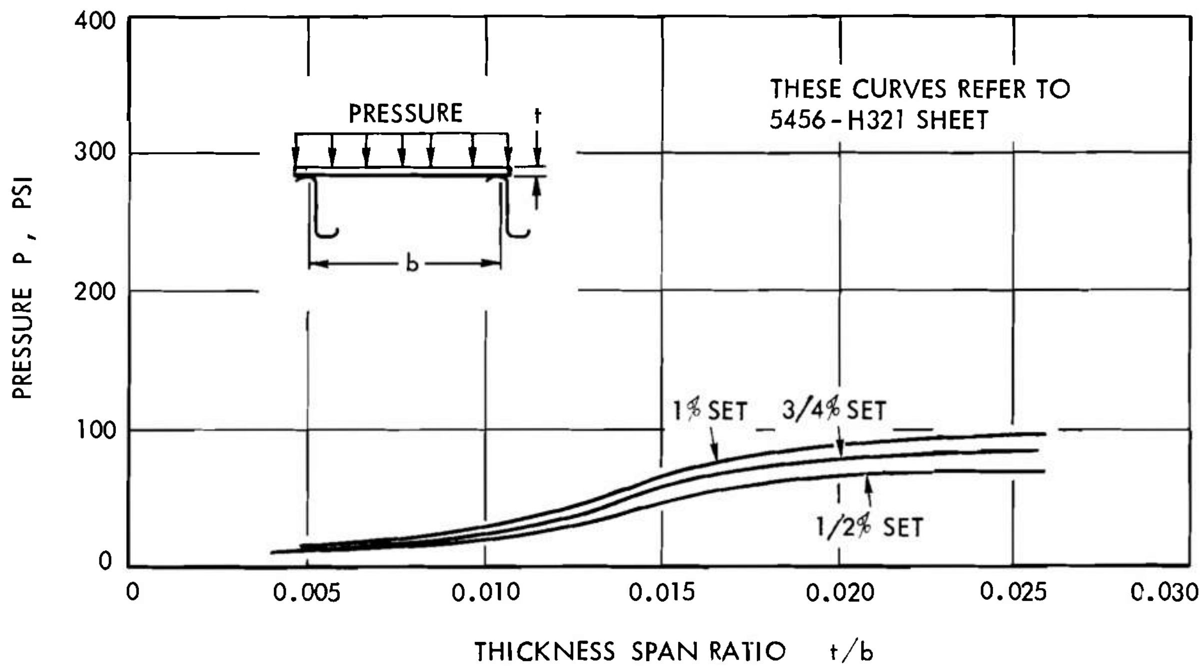


Figure 7-36. Bottom Plating Deformation Curves

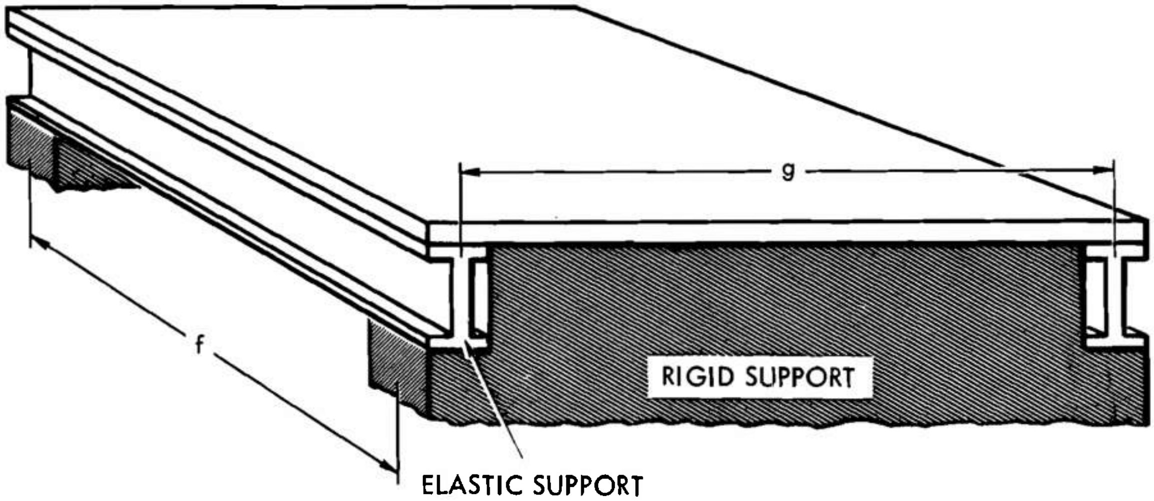


Figure 7-37. Rectangular Flat Plate With Two Edges on Elastic Support

7-3.3.2.3 Plate and Stiffeners

Almost all large stressed skin structures are built with plate and stiffener construction. In general the weight of the structure decreases as the spacing between stiffeners decreases so that monocoque is the heaviest configuration and sandwich, which represents a structure with the distance between stiffeners reduced to zero, is the lightest. The problems inherent with sandwich construction have limited the use of that configuration but it does have promising potential and can be used if the extra construction difficulty is worth the reduction in weight.

The general solution to designing a lightweight structure is to reduce the stiffener spacing and the skin thickness to the minimum practical with the stiffener depth as large as is necessary. The minimum distance or thicknesses are related to the problems of construction, repair, local or abuse loads, and cost. As the scantlings get lighter and the pieces smaller, the cost of handling each piece goes down but the number of pieces goes up so that the total cost is higher. Part of the project management is to determine the "value" of reducing the weight.

There are an unlimited number of potential stiffener shapes that might be used. The ones that are popular are listed in Table 7-19.

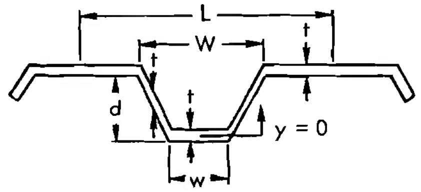
The flat bar stiffeners are used where the loading is low and where they are sufficiently strong in the minimum thickness plate used. In some cases where the shell is in sheets or plates, the thickness of the shell will be standardized even though the loads are variable. In the more lightly loaded areas, flat bar stiffeners will be used. They are vulnerable to damage from lateral abuse loads at the upper edge. They can be installed with a bulb on the upper edge to resist these loads.

If mechanical fasteners or spot welding are used, the stiffener must have a faying flange against the plate. In addition to this function, the faying flange reduces the effective distance between stiffeners so that fewer stiffeners may be required. In general when the stiffeners are either welded or extruded integral with the plate, the use of a faying flange will result in a higher weight.

The inverted angle is convenient to use since the orientation of the outstanding flange can be alternated and the backs can then be fastened together with less fitting problems.

The tee is widely used because it offers the lightest total weight with a balanced section. It is the shape most frequently extruded integrally with the plate. The extrusion procedure is, of course, limited to aluminum and the primary advantage is the reduction of welding and the accompanying distortion. Table 7-20 shows ALCOA's tentative limits of integrally stiffened extrusions.

Flanged plates are frequently used where deep stiffeners are required, primarily for indirect framing. On the assumption that the structural



PRESSED CORRUGATION
(MODEL USED TO DERIVE EQUATIONS)

A = PARAMETER DEPENDING ON K_2 AND K_3

d = DEPTH OF CORRUGATION, IN.

I = MOMENT OF INERTIA, $IN.^4$

K_1 = RATIO d/t

K_2 = RATIO w/W

K_3 = RATIO W/L

L = CORRUGATION PITCH LENGTH, IN.

t = THICKNESS OF SHEET, IN.

w = WIDTH OF BOTTOM OF CORRUGATION, IN.

W = WIDTH OF TOP OF CORRUGATION, IN.

$\bar{y}$ = HEIGHT OF NEUTRAL AXIS, IN.

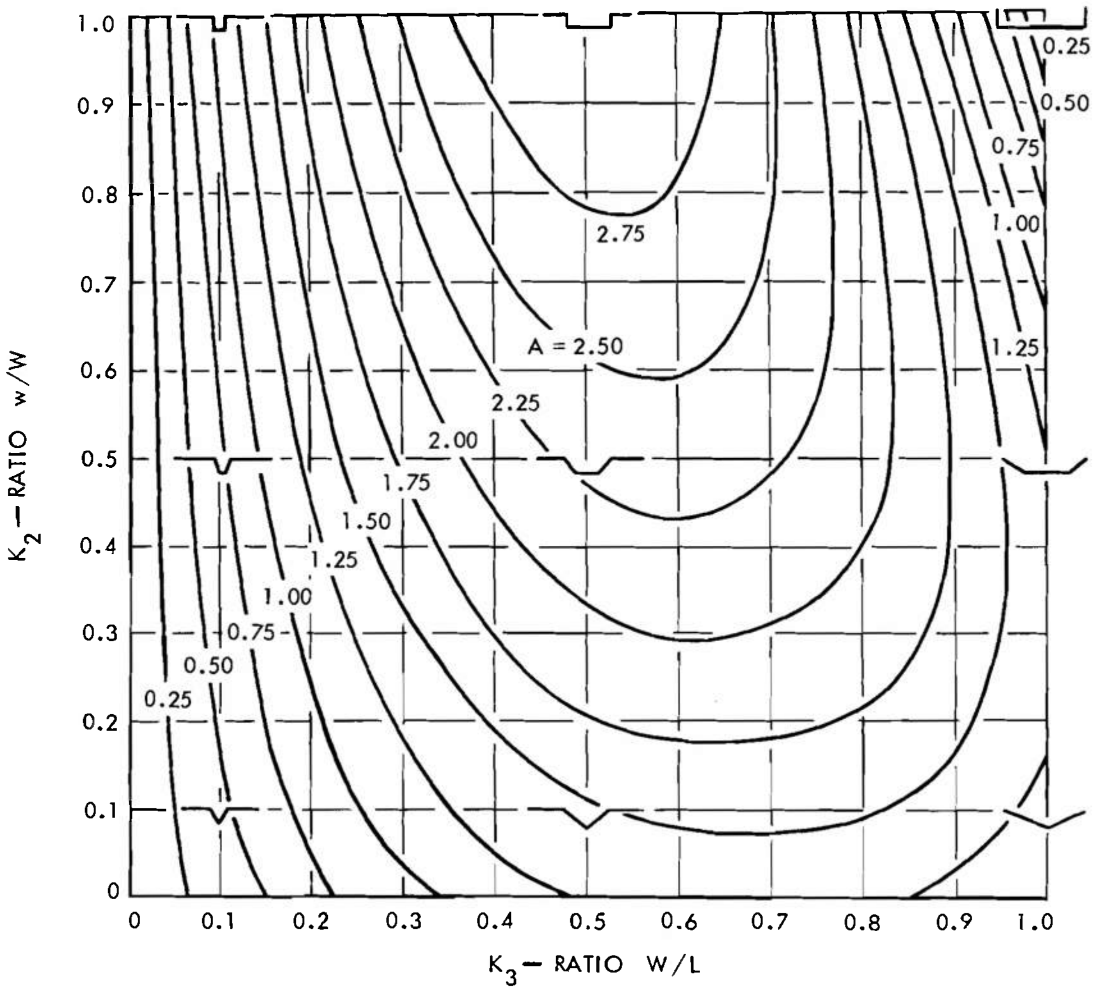
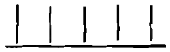
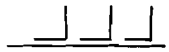
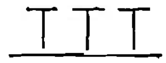
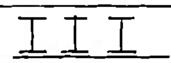


Figure 7-38. Nomenclature and Plot of Constant "A" for Corrugated Material

TABLE 7-19 STIFFENER SHAPES IN GENERAL USE

Shape	*Cross Section*	*Remarks*
Flat Bar		Simple but unstable at high loads.
Angle		Mechanical fasteners or spot welded construction.
Inverted Angle		Simple but unbalanced.
Inverted Tee		Most widely used. Balanced construction.
Tee		Mechanical fasteners or spot welded construction—spreads the load on the plate.
Channel		Mechanical fasteners or spot welded construction. Formed from sheet, extruded or rolled. Slightly unbalanced.
Inverted Channel		Used for aluminum offshore race boats. Interior cannot be maintained. Reduces effective distance between stiffeners.
Zee		Widely used in aircraft industry. Mechanical fasteners or spot welded.
I or H		Mechanical fasteners or spot welded. Balanced.
Flanged Plate		Versatile for materials with a short bend radius.

arrangement and the plating thickness have been selected, the adequacy of the stiffeners can be checked. Some or all of the shell will act as a flange for the stiffener. There is a phenomenon known as shear-lag which limits the stiffening effectiveness of plating attached to a stiffener. Because of the shear strain, the longitudinal tensile or compressive stress in wide beam flanges diminishes with the distance from the web. Ref. 36 presents the results of several investigators on the problem of shear lag. It depends not only on the method of loading and support, and the ratio of span to flange width, but also on the ratio of modulus of rigidity to the modulus of elasticity and on the ratio:

$$\frac{(3I_w + I_f)}{I_w + I_f}$$

where

I_w = moment of inertia about the neutral axis with respect to the web, in^4

I_f = moment of inertia about the neutral axis with respect to the flanges, in^4.

The Navy (Ref. 28) uses a maximum effective breadth of plating for the calculation of section modulus of

$$2t\sqrt{E/\sigma_y}\text{ , in.}$$

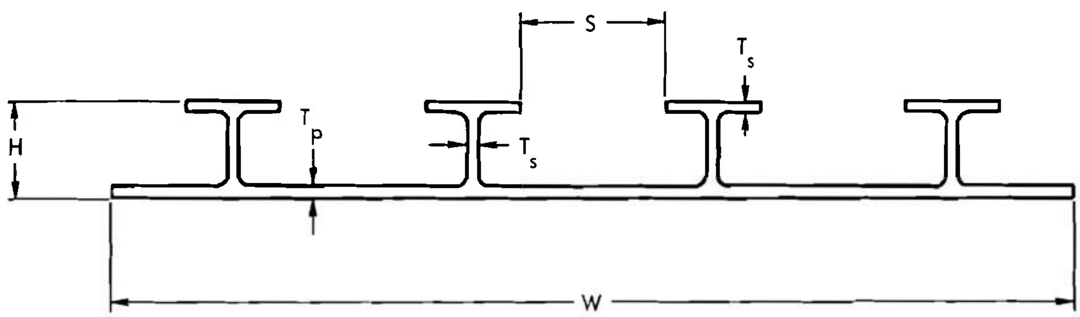
where

t = plate thickness, in.
E = modulus of elasticity, psi
σ_y = yield strength, psi

The maximum effective breadth is given in Table 7-21 for the more common materials.

The section moduli and cross-sectional areas of the stiffener and plating combination can be calculated using the standard method (Ref. 36). For body-bound mechanical fasteners, the net section should be used for tension and the gross section for compression. By use of these calculated section characteristics and the loads, the stresses and stability can be calculated. Table 7-21 contains a variety of formulas that can be used for these calculations.

TABLE 7-20 LIMITS OF WIDE INTEGRALLY STIFFENED EXTRUSIONS



ALLOYS	MAXIMUM W	MINIMUM T_p	MINIMUM T_s	MAXIMUM H*	MINIMUM S
5083, 5086	21.000	0.156	0.125	2.0	2.0
	26.000	0.188	0.156	2.0	2.0
	32.000	0.250	0.188	2.0	2.0
	36.000	0.375	0.250	2.0	2.0
5456	21.000	0.156	0.125	2.0	2.0
	26.000	0.188	0.156	2.0	2.0
	32.000	0.250	0.188	2.0	2.0
	36.000	0.500	0.375	2.0	2.0
6061	23.375	0.094	0.078	2.0	2.0
	28.675	0.094	0.078	2.0	2.0
	34.313	0.094	0.078	2.0	2.0
	39.000	0.188	0.125	2.0	2.0

* THIS HEIGHT BASED ON MINIMUM THICKNESS
GREATER HEIGHTS ARE POSSIBLE WITH HEAVIER SECTIONS
ALL DIMENSIONS IN IN.

TABLE 7-21 COMBINED STIFFENER ON PLATE FORMULAS (REF. 28)

b_f - Breadth of outstanding flange of a balanced stiffener

d - Depth of stiffener

E - Modulus of elasticity

f_b - Compressive stress from bending load

f_c - Compressive stress from axial load

f_d - Compressive stress in plating from loads normal to plating

F_c - Stress at failure from axial loading

F_u - Stress at failure from compressive loading

F_y - Tensile yield stress

t - Plating thickness, in.

t_f - Flange thickness

t_w - Web thickness

(1) To calculate section modulus use a maximum effective breadth of plating equal to the stiffener spacing or

$$2t\sqrt{\frac{E}{F_y}} \quad \text{whichever is less}$$

Material	$2t\sqrt{\frac{E}{F_y}}$
Mild Steel	60 t
Hi Tensile Steel	51 t
Al 5456-H321 parent	35 t
Al 5456-H321 welded	40 t

TABLE 7-21 (Concluded)

(2) Combined bending and compression interaction formula

$$\frac{f_c}{F_c} + \frac{f_b}{F_y} \leq 1$$

(3) Breadth/thickness ratios for balanced flanges should be equal or less than

$$\sqrt{\frac{E}{F_y}}$$

Material		Max b/t
Mild Steel		30
Hi Tensile Steel		26
Al 5456-H321	parent	18
	welded	20

(4) Maximum ratio of span L to flange width b_f, should equal or be less than

$$\frac{1.283\sqrt{E/F_y}}{\sqrt{1 + 0.333\,\frac{dt_w}{b_f t_f} - \frac{E}{7.8F_y}\left(\frac{t_f}{d}\right)^2}}$$

(5) Ratio of stress at failure to stress under specified conditions of loading for stiffened panels with compressive and normal loading is

$$\frac{1}{\left(\frac{f_c}{F_u} \times \frac{F_y}{F_c}\right) + \frac{f_d}{F_y}}$$

7-3.3.3 Indirect Framing

The primary use of the indirect framing is to support the shell and direct framing. It includes athwartship and longitudinal bulkheads and girders. These members also may provide watertight compartments and machinery foundations. Included with this group, although not necessarily fitting the description, are stanchions. The indirect framing usually is required to carry the bending moment between adjacent side and deck or side and bottom. Transverse bulkheads do this with a shear load, and transverse girders act as a ring frame. Frequently brackets are fitted between deck and side, or bottom and side girders, to carry the bending moment.

The design of the shell and direct framing furnish the loads that must be carried by the indirect framing. The transverse girders should be analyzed as ring frames (Ref. 35); the longitudinal girders, as beams. The bulkheads can be considered to be short, deep plate girders although the loads are almost entirely shear. Access is a critical concern when designing indirect framing because of the depth of the structure involved. Ref. 40 presents formulas that can be used for the design. The analysis involved for the bulkheads, in particular, is difficult and careful consideration of past practice may be helpful.

7-3.3.4 Ramps

The structural design of ramps must consider three problem areas:

1. Deck loads
2. Ground support loads
3. Hinges

The deck or roadway is subject to dynamic loads from vehicles and from cargo. The critical loads are likely to result from the grousers on tracked vehicles or the wheels on loaded fork lift trucks. It is sometimes advantageous to place the stiffeners on top of the plate where they can serve a secondary function as tread bars for wheeled vehicles.

In the lowered position, the ramp is supported at the after end by the hinges and at the forward end by the ground. The ramp and the hinges must be constructed to resist the torsion that occurs when the ground is uneven and the ramp is supported only on one side at the forward end. Also the forward side must be able to resist the local loads when it is supported by a rock or other small hard object.

7-3.3.5 Miscellaneous

There are many miscellaneous fittings that apply a load over a small area. These include:

a. Cleats
b. Foundations
c. Lifting pads
d. Cargo tie-down fittings
e. Appendages

The manner in which the stress is dissipated must be carefully considered because large forces are involved and the results of a failure can be serious. For instance, consider the results of a lifting pad failure while the amphibian is being unloaded from a cargo ship and it is 40 ft above the bottom of the hold.

There are standard patterns of cleats available so that the cleat itself is not a design problem. The base of the cleat, however, must be fastened such that it can take the impact loads from the strongest rope that will fit on the cleat.

Foundations can add much extra, unplanned weight if they are not carefully designed. The propeller thrust must be carried by either the transmission or the engine foundation, depending upon the design. Vibration effects must be considered. Some previous vehicles have had trouble when the fixed ends of the hydraulic wheel steering cylinder were fastened to the wheel well plate without a special foundation plate. All places where a concentrated load is applied should be specially considered.

Lifting pads should be designed to meet the test specifications. Some of the common tests are shown in Fig. 8-2. The pads must fit the standard lifting gear and they must be able to take the side loads. It should be noted that the pads are convenient for other tasks besides lifting.

Cargo tie-downs must be designed to take standard fittings and at a wide range of pull angles. There have been problems in the past with water and dirt collecting in the tie-down recesses and freezing which has made them difficult to use.

The appendages including the rudder and the propeller are damaged occasionally, particularly in the land mode. It is important that they fail

before the connection to the hull so that the watertight integrity of the hull is not broken even though some component is not working.

7-3.4 DETAIL DESIGN

During the preliminary design, the major decisions for the structural characteristics are made. These include, basically, the material, configuration, hull shape, and smoothness. Along with these selections is a series of goals which may or may not be promulgated as such. These include structural weight, position of the center of gravity, hull cost, safety level, and maintenance and repair levels. Additional goals which should be known during detail design are potential manufacturers, anticipated development programs, hull tests, approximate vehicle quantity, and production schedule.

As mentioned previously, communications between designers on a project must not deteriorate at this phase due to the rush to finish the design. The final structural design has a major impact on the total design. The probability of meeting, surpassing, or missing the goals must be made known to the project manager. The structural designer must appreciate the real mission of the user of the vehicle. This mission at times demands the maximum performance from the vehicle, regardless of the original mission for which the vehicle was designed. It is the responsibility of the designers to produce a design that is as safe as practical for this condition and not overlook some detail that will cause a catastrophic failure.

7-3.4.1 Factors of Safety

The rational process of design of each load resisting member may be considered in four steps which are:

1. Determine mode of failure.
2. Determine a relationship between the loads and the quantity which is most significant in the failure of the member.
3. By appropriate tests of the material used in the member, determine the maximum value of the quantity that is considered to cause structural damage.
4. By use of experimental observations and analyzing experience, judgment, and the basic design philosophy for the project; select a value of the quantity for use in Step 2 which will allow the member to fulfill its duty as required, in spite of human ignorance or error. Normally Step 2 is, in some way, an idealized situation of the actual service conditions of the member. In many cases the loads cannot be predicted accurately; the material is not completely uniform; the fabrication is not the best quality; the material will deteriorate in service; the relationship in Step 2 is only an approximation using simplifying assumptions; or many other conditions exist which reduce the level at which failure will occur. Step 4, then, is used to apply a reduction factor, to cover all of the conditions and considerations that cannot be expressed quantitatively in mathematical form, and applied in Step 2.

The most common reduction factor is the factor of safety which is the ratio of the load that would cause failure to the load that is imposed in service. It is not a ratio of stresses. For those situations where the load is proportional to the quantity that first causes structural damage, the factor of safety is the same as the ratio of stresses or other quantity of failure. Table 7-22 lists typical factors of safety and the values on which they are based. For steel structures the effects of corrosion should be countered by the addition of a uniform increase in the scantling, normally called a "corrosion allowance".

7-3.4.2 Fasteners

A chain of structural members is only as strong as the weakest member. Fasteners are one of the links in the chain and they must be designed with as much care as the members. The three main types of fasteners are:

1. Mechanical
2. Welding
3. Adhesive

7-3.4.2.1 Mechanical

Mechanical fasteners can be further subdivided into

a. Bolts
b. Rivets
c. Special purpose (primarily patented types)

Although the actual stress patterns in a bolt or riveted joint are complex, it is customary to make certain simplifying assumptions and apply certain factors of safety. The system (Ref. 40)

TABLE 7-22 TYPICAL FACTORS OF SAFETY

Type of Structure	Ref.	Factor of Safety	Strength Basis	Material	Remarks
Main Engine Foundation LARC V	47	2.0	Ultimate	Aluminum	Re-design after failure
LVPT Sheet and Plate Shapes	46	1.46 1.28	Yield Yield	Aluminum	Study
LVK-Structure	47	1.1 1.5	Yield Ultimate	Aluminum	Proposal
LVH-X2 Structure	48	1.15 1.5	Yield Ultimate	Aluminum	A carry-over factor of 1.1 used so that foundations fail before hull structure
Bridge-Ferry	49	Tension 1.25 Compression 1.43 Shear 1.54 Pina-flexure 1.11 Bearing 1.43	Minimum Yield	Aluminum	Specification Fatigue stress, in general use not more than 1/2 of yield
Planing craft	37	1.1	Yield	Aluminum	Excludes plating for which 0.005 permanent set is suggested
Bridges	40	2.2 1.85 1.35	Ultimate Yield Incipient buckling	Aluminum	ASCE Specifications Allowances of percent static loads for dynamic loads
Buildings		1.95 1.65	Ultimate Yield		Elevator support 100 Traveling crane supports 25
		1.35	Incipient Buckling		Light machinery supports 20 Reciprocating machinery supports 50
Hydrofoil Structure		1.0 1.5	Fatigue with a required life Ultimate	Aluminum	Shell designed for a (0.005 x span) permanent set
Deck		1.84	Compression Yield		Deflection criteria $EI/(span)^3 \geq 60$
Ships	50	1.22 1.18 1.45 1.22 1.15 1.08 1.24 1.24 1.14 1.14 1.52	Yield Yield Yield Yield-HAZ " " " " " " Yield-parent	Mild Steel HTS HY-80 5086-H32 5086-H111 5086-O 5456-H321 5456-H111 5456-H34 5454-H111 6061-T6	
Boats	29	2 4 4 6 6 10	Ultimate " " " " "	Fiberglas	Deflection limit = 1/200 span Static short term loads Static long term loads Variable or changing loads Repeated loads Fatigue or load reversal Impact loads-repeated
Minesweeper hull		4 2	Ultimate Ultimate	Fiberglas	Allows for unknowns in estimating underwater explosion pressures Panel buckling deflection limit = 1/150 span.

which follows is used in the aerospace industry which depends heavily on mechanical fasteners. The assumptions normally made are:

a. The applied load is carried entirely by the fastener. Friction between the connected parts is ignored.

b. When the center of gravity of the fasteners is on the assumed line of action of the load, the fasteners are all loaded in proportion to their respective section areas.

c. Shear stress is uniformly distributed across the fastener section.

d. The bearing stress between plate and fastener is uniformly distributed over an area equal to the fastener diameter times the plate thickness.

e. The stress in a tension member is uniformly distributed over the net area.

f. The stress in a compression member is uniformly distributed over the gross area.

The possibilities of failure due to secondary causes, such as shearing or tearing out of a plate between fastener and edge of plate, or between adjacent fasteners, are minimized by using the following standards:

a. Edge distance from a fastener to a sheared edge shall not be less than 1¾ diameters, or to a rolled edge 1½ diameters.

b. The minimum fastener spacing is 3 diameters.

c. The maximum fastener pitch in the direction of stress shall be 7 diameters; at ends of a compression member 4 diameters or a distance equal to 1½ times the width of the member (for stiffeners).

d. For a zig-zag joint the net width of the part shall be obtained by deducting from the gross width the sum of the diameters of all the holes, and adding for each gage space $S^2/4g$, where S = longitudinal spacing of any two successive holes and g = spacing of the same two holes transverse to the direction of stress.

e. The shear and bearing stresses shall be calculated on the basis of the nominal fastener diameter; the tensile stresses on the hole diameter.

f. Special consideration must be given to long fasteners (grip length/diameter greater than 3).

g. For bolts in tension the factor of safety must be at least 1.5. The use of aluminum bolts in tension should be avoided.

h. Hole-filling fasteners (rivets) should not be combined with nonhole-filling fasteners (bolts). The shear and bearing failure loads must be separately calculated and the lower value used.

The SAE Standard J429 recognizes the 9 grades of bolts shown in Table 7-23. For combined loads of shear and tension the interaction formula is on Fig. 7-39. Clearances required to install bolts are shown on Fig. 7-40. Unit bearing strengths are given in Figs. 7-41 and 7-42. For the features and design procedures for special fasteners, the manufacturer should be consulted.

7-3.4.2.2 Welding

Welding is the primary method of attachment for the present fleets of amphibians. One of the primary reasons for using welding is that mechanical fasteners in aluminum usually leak after the joint has been severely stressed a few times. Largely all-welded ships have been the standard since World War II, and the proper procedure for welding steel is widely known. Butt welds, when properly back chipped to remove all oxidized and unfused metal, are as strong as the parent metal. For amphibians, all butt welds should be designed for full penetration and therefore full efficiency. Butt welds may be laid out to suit fabrication and need not be staggered or offset.

The efficiency of fillet-welded connections can be calculated but it is more convenient to use safe design loads. Table 7-24 lists the allowable design loads for various fillet sizes in mild steel. These loads are based on an allowable shear value of 13,600 psi on the throat of the fillet and are taken from Ref. 41. These are safe loads for design, the breaking loads are about 4 times the values tabulated. Intermittent fillet welds are generally sufficient but where a double continuous fillet weld is desired, such as for high impact loads, the fillet should be 0.707 times the thickness of the thinner plate. Table 7-25 from Ref. 41 lists the fatigue strengths of various types of joints for ASTM-A7 structural steel. Note the poor fatigue characteristics of the tee joints in the fillet weld, and the plug and slot welds. In general any joint that can be seen can be welded, but additional clearance will reduce the cost and increase the quality of the weld.

Welding aluminum successfully requires a higher quality of workmanship than for steel. There have been major advances in the knowledge, techniques, and equipment for

TABLE 7-23 SAE BOLT GRADES

Grade	Bolt Size Dia, in.	Proof Load, psi	Tensile Strength, min, psi	Hardness	
				Brinell	Rockwell
0	All sizes	-	-	-	-
1	All sizes	-	55,000	207 max	95 B max
2	Up to 1/2 incl over 1/2 to 3/4 incl	55,000 52,000	69,000 64,000	241 max	100 B max
	Over 3/4 to 1-1/2 incl	28,000	55,000	207 max	-
3	Up to 1/2 incl Over 1/2 to 5/8 incl	85,000 80,000	110,000 100,000	207-269	95-104 B
5	Up to 3/4 incl Over 3/4 to 1 incl Over 1 to 1-1/2 incl	85,000 78,000 74,000	120,000 115,000 105,000	241-302 235-302 223-285	23-32 C 22-32 C 19-30 C
6	Up to 5/8 incl Over 5/8 to 3/4 incl	110,000 105,000	140,000 133,000	285-331 269-331	30-36 C 28-36 C
7	Up to 1-1/2 incl	105,000	133,000	269-321	28-34 C
8	Up to 1-1/2 incl	120,000	150,000	302-352	32-38 C

aluminum welding in the last few years and some manufacturers are apt to be using out-dated techniques. Several characteristics of aluminum affecting the welding process are:

a. Aluminum melts at about 1200° F and does not become "red-hot" or otherwise change color when heated. The oxides of aluminum melt at about 3600°F or some 2400°F higher than the parent metal.

b. Aluminum conducts heat 3 times faster than steel and the thermal expansion is approximately twice as great. Distortion from welding must be carefully controlled.

c. Cleaning the weld surface of oxide, grease, or other foreign material just prior to welding is necessary to produce good welds with low porosity. The lower melting temperature, lower specific gravity, and higher heat conduction are some of the reasons why this problem is much worse for aluminum than for steel.

To minimize distortion, it is necessary to use:

a. Minimum amount of weld material

b. Back up chill bars to absorb heat

c. Rigid jigs

d. A high speed, concentrated heat input welding process

e. Equalized welding on both sides of material

f. Proper welding sequence

g. Increased scantlings

As with all types of fabrication, some residual stress will be built into any redundant structure. For welded aluminum this problem is compounded by the large welding distortion. The major method of combating residual stress is proper weld sequence.

Ref. 42 is addressed to high quality welding for the construction of missiles. There are, however, two items of special interest in this reference:

a. Very high quality welds are practical when the material is tack welded to hold the alignment. The general practice has been to avoid tack welds, but they can be used if properly cleaned prior to production welding.

b. Fiberglas tape with adhesive backing can be used as a backup in lieu of metal backup bars, without weld contamination.

Ref. 43 is an extensive collection of information on welding aluminum. It covers all types of welding including:

a. Arc (MIG and TIG)

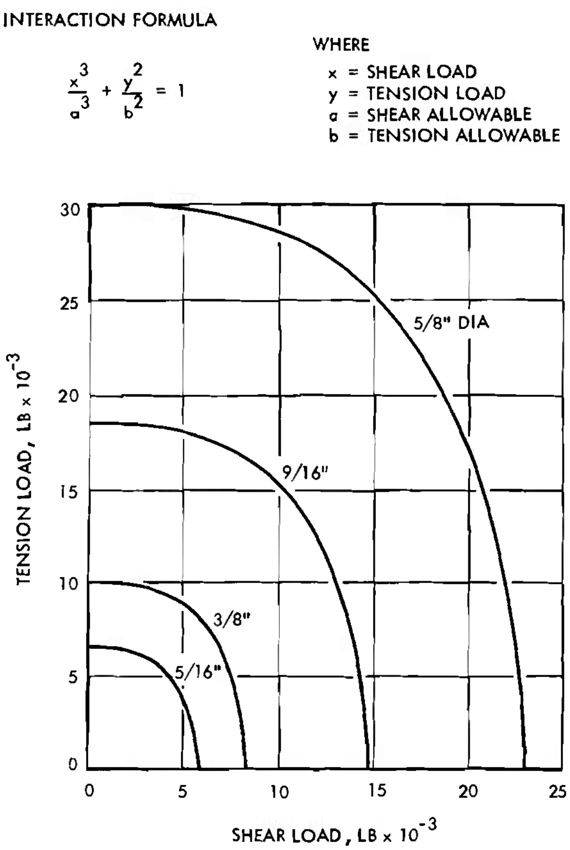


Figure 7-39. Combined Shear Load and Tension Load on Steel Bolts (With an Ultimate Tensile Strength of 125,000 psi)

b. Resistance (seam, spot, flash butt)
c. Gas
d. High frequency
e. Pressure
f. Ultrasonic
g. Electron beam
h. Laser

Strengths of butt joints are listed in Table 7-26.

The stress in a double fillet weld in longitudinal shear is:

$$\sigma_s = \frac{F}{1.414sL} \tag{7-43}$$

where

σ_s = shear stress in weld, psi
F = total longitudinal force, lb
s = fillet size (length of fillet leg), in.
L = length of weld, in.

Fillet welds are loaded in transverse shear if the load is applied perpendicular to the axis of the weld. For equal leg length transverse fillet loading, the plane of maximum strength is not at the minimum throat depth but, theoretically, is at an angle of 22½ deg with the fillet leg for the maximum shear stress, and at an angle of 67½ deg for the maximum tensile stress. The stress will be the same in both of these planes so the weld will fail in shear since the ultimate shear strength is lower than the ultimate tensile strength. Table 7-27 lists filler alloy shear strengths. The stress values produced by the longitudinal shear loading are always higher than the shear stresses from an identical transverse loading. It is the practice of some designers in fillet weld design to assume the longitudinal shear loading regardless of the direction of the applied load. Fig. 7-43 shows the transverse shear strength of alloy 5356 and 5556. The failure is assumed to be in the fillet.

A common application of fillet welds is for fully connecting two members at right angles, using a double fillet weld. If it is desired to make this welded connection strong enough so that in severe loading failure would occur in the base metal of the web, then the three methods of failure in the weld area are:

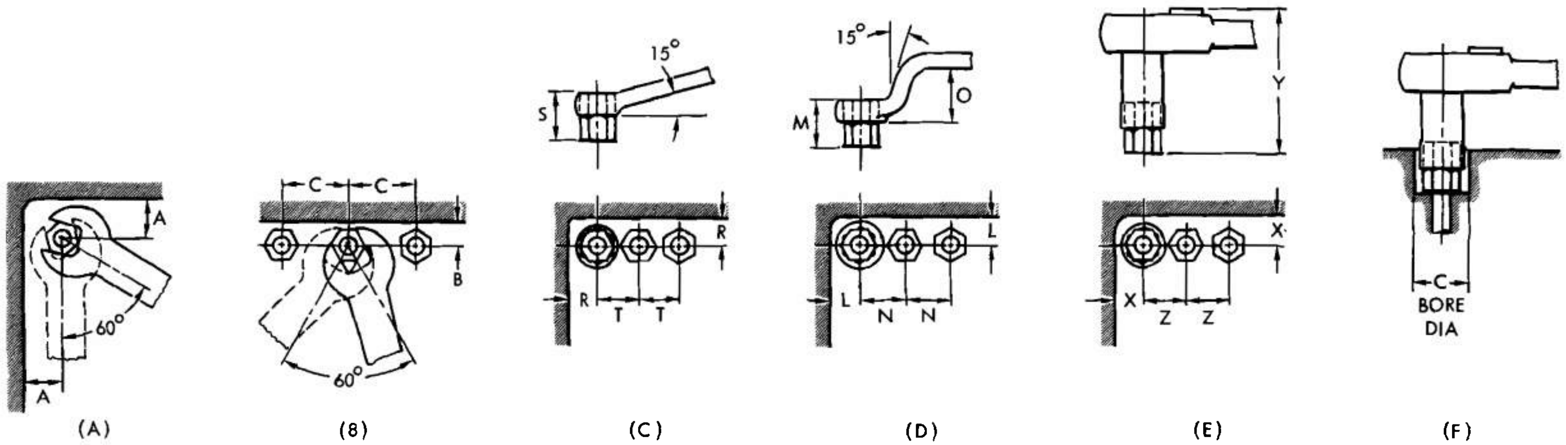
1. Failure in a plane through the fillets at or near the minimum throat depth
2. Failure in an irregular plane that roughly follows the fusion line of the fillet
3. Transverse tensile failure in web

Figure 7-44 illustrates these assumptions. The length of the irregular shear plane q, by measurement is about 1.1 times the length of the fillet leg.

The following items govern the fillet size and the strength of the weld metal:

a. The fillets must be sufficiently large to provide an adequate length of shear plane q adjacent to the fillets so that failure does not occur as base-metal shear adjacent to the fillets. It is general practice to assume the heat affected zone (HAZ) extends one inch into the parent metal from either the center of a butt joint or the heel of a fillet, regardless of the weld size.

b. After the minimum fillet size is determined by the length of shear plane q, a corresponding minimum fillet weld shear strength must be established so that fillet failure does not occur before base metal failure in the web.



BOLT SIZE NOMINAL	BOLT HEAD OR NUT ACROSS FLATS	AN8505 STYLE OPEN END WRENCH (FIGURES A,B) CLEARANCE CORNER MIN A	CLEARANCE ROW MIN B	MIN BOLT SPACING C	BOX WRENCH - 15° ANGLE (FIGURE C) CLEARANCE ROW & CORNER R	CLEARANCE HEAD MIN S	MIN BOLT SPACING T	OFFSET BOX WRENCH (FIGURE D) CLEARANCE ROW & CORNER L	CLEARANCE HEAD MIN M	MIN BOLT SPACING N	OFFSET O	SOCKET WRENCH (FIGURE E) CLEARANCE ROW & CORNER X	CLEARANCE HEAD MIN Y	MIN BOLT SPACING Z	MIN "C" BORE DIA BASED ON SOCKET END (FIG.F)
No. 8	0.344											13/32	1-15/16	19/32	5/8
No. 10	0.375	7/16	3/8	11/16	11/32	21/32	9/16					1/2	2- 3/4	23/32	11/16
1/4	0.438	1/2	15/32	13/16	3/8	3/4	5/8	13/32	25/32	21/32	13/16	1/2	2-13/16	3/4	3/4
5/16	0.500	1/2	1/2	27/32	13/32	27/32	11/16	7/16	27/32	3/4	7/8	1/2	2-27/32	25/32	7/8
3/8	0.563	5/8	9/16	1	15/32	31/32	25/32	1/2	31/32	27/32	15/16	1/2	2-15/16	27/32	15/16
7/16	0.625	5/8	19/32	1 - 1/32	1/2	1 - 1/16	27/32	17/32	1 - 1/32	15/16	1	1/2	2-31/32	7/8	1 - 1/16
1/2	0.750	23/32	23/32	1 - 3/16	17/32	1 - 1/8	1	5/8	1 - 7/32	1 - 3/32	1 - 1/16	19/32	3 - 3/32	1	1 - 3/16
9/16	0.875	3/4	11/16	1-11/32	21/32	1 - 3/8	1 - 5/32	11/16	1 - 5/16	1 - 7/32	1 - 3/16	25/32	4 - 1/16	1 - 9/32	1 - 3/8
5/8	1.000	27/32	3/4	1 - 1/2	25/32	1-19/32	1 - 3/8	13/16	1-17/32	1 - 3/8	1 - 5/16	13/16	4 - 5/32	1 - 3/8	1 - 5/8
3/4	1.125	15/16	27/32	1 - 3/4	31/32	1-25/32	1-19/32	1	1-21/32	1-21/32	1 - 3/8	7/8	4 - 3/8	1-19/32	1 - 3/4
7/8	1.313	1 - 1/8	1	2	1 - 1/16	1-31/32	1-27/32	1 - 1/32	1-25/32	1-25/32	1 - 3/8	1	4 - 5/8	1-13/16	2 - 1/16
1	1.500	1 - 7/32	1 - 1/32	2 - 3/16	1 - 3/32	2 - 3/32	1-31/32					1 - 3/32	4 - 7/8	2 - 1/16	2 - 1/4

Figure 7-40. Minimum Wrench Clearances

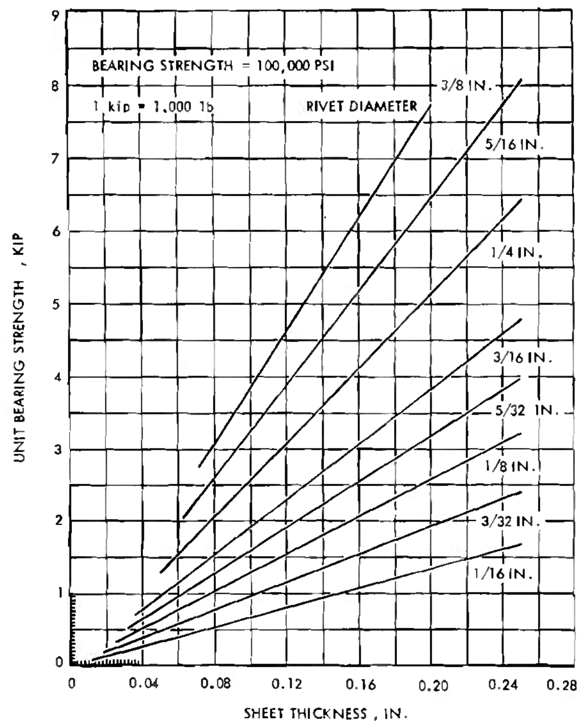


Figure 7-41. Unit Bearing Strength of Sheets on Rivets

c. If a weld metal weaker than the base metal is used, the fillet size must be increased accordingly.

d. There is little benefit in using an overly strong weld metal with the object of reducing fillet size. If this is done, the base metal shear strength through the irregular plane q at the fillet fusion line will not have sufficient strength.

In applying calculations to fulfill Item a, the given symbols for a one-inch-long segment of the structure are:

The unit length web tensile strength

$$W_{tension} = tB\text{, lb} \tag{7-44}$$

The unit length shear strength

$$W_{shear} = tS_1\text{, lb} \tag{7-45}$$

In shear at the weld fusion lines, the unit length shear strength

$$W_{fusion\ lines} = 2qS_2\text{, lb} \tag{7-46}$$

where

For the Web Material

t = web thickness, in.

B = web material tensile ultimate strength, psi

S_1 = web material shear strength, psi

In the Weld Area

q = length of shear plane, in.

S_2 = weld material shear strength, psi

2 = number of fillets

When the welds are loaded in transverse shear, the web is in tension. To insure that failure will not occur in web shear at the fusion line, the web-shear strength on plane q near the fillets must at least equal the web tensile strength, thus:

$$W_{tension} = W_{fusion\ line}$$

$$tB = 2qS_2$$

with $S = S_1 = S_2$

$$q = \frac{tB}{2S} \text{ but } q = 1.1s\text{, where s is the fillet size} \tag{7-47}$$

$$1.1s = \frac{tB}{2S}\text{ ; } s = \frac{tB}{2.2S} = \text{minimum fullet size, when the welds are loaded in transverse shear.}$$

When the welds are loaded in longitudinal shear, the web is also in shear. In this case let $W_{shear} = W_{fusion\ line}$

$$tS_1 = 2qS_2$$

$$q = \frac{tS_1}{2S_2} = \frac{t}{2}\text{, but } q = 1.1s\text{, so } 1.1s = \frac{t}{2} \tag{7-48}$$

$$s = \frac{t}{2.2} = \text{minimum fillet size when the welds are loaded in longitudinal shear.}$$

The foregoing calculations show how the shear strength of the web base metal near the fillet can determine the minimum fillet size. After the minimum fillet size is determined, the minimum strength of the filler metal must be determined in fulfillment of Item b relative to fillet size and strength. For a one-inch-long segment of the structure, the fillet weld unit length shear strengths are:

$$\text{Transverse shear strength} = 0.7071sT\text{, lb} \tag{7-49}$$

$$\text{Longitudinal shear strength} = 0.7071sL\text{, lb} \tag{7-50}$$

and the fusion line unit length shear strength is:

$$W_{fusion\ line} = 1.1sS\text{, lb} \tag{7-51}$$

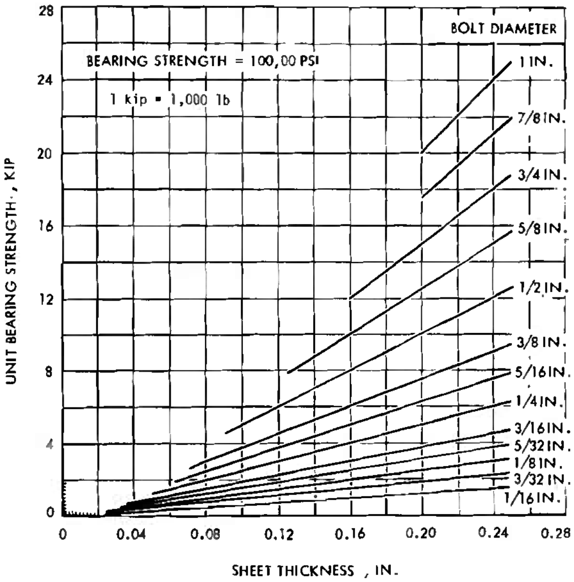


Figure 7-42. Unit Bearing Strength of Sheets on Bolts

where

In the Fillets

0.7071s = weld failure path, in.
T = transverse shear strength, psi
L = longitudinal shear strength, psi

In Fusion Line Shear

q = 1.1s = fusion line failure path, in.
S = shear strength of base metal, psi

By equating the fillet weld shear strength to the fusion line shear strength, the minimum filler metal shear strength can be found, for the minimum size fillet.

Transverse

$$0.7071sT = 1.1sS$$

$$T = \frac{1.1S}{0.7071} \tag{7-52}$$

$$T = 1.555S$$

Longitudinal

$$0.7071sL = 1.1sS$$

$$L = \frac{1.1S}{0.7071} \tag{7-53}$$

$$L = 1.555S$$

TABLE 7-24 ALLOWABLE LOAD ON FILLETS (13,600 psi SHEAR ON THROAT)

Size, in.	*Allowable load per inch of length, lb*
3/16	1800
1/4	2400
5/16	3000
3/8	3600
7/16	4200
1/2	4800
5/8	6000
3/4	7200
7/8	8400
1	9600

If the minimum fillet size is used, the filler metal must have at least 1.555 times the base metal shear strength to avoid shear failure in the weld. Thus

$$\frac{T}{S} \text{ or } \frac{L}{S} = 1.555.$$

If this ratio is not met, the minimum fillet size must be increased by a ratio of

$$\frac{1.555}{T/S} \text{ or } \frac{1.555}{L/S},$$

which fulfills Item c of the precepts governing calculations for fillet size.

In accordance with Item d, a filler metal shear strength which meets the requirement of 1.555 times the base metal shear strength is adequate. There is no advantage in using stronger filler metal because the minimum fillet size is established by the base metal shear strength.

Joint accessibility is an important factor to consider. It is very easy to design a part which cannot be welded because part of the joint is not accessible to the welding tip. Typical MIG guns with dimensions are shown on Fig. 7-33. A full size template of the gun to be used is useful.

7-3.4.2.3 Adhesives

Ref. 44 discusses adhesive aluminum bonding, including:

TABLE 7-25 FATIGUE STRENGTH OF STEEL WELDS, psi x 10^{-3}

ASTM-A7 Structural Steel

Type of Joint	0 to Tension		Reversed	
	Cycles 100,000	Cycles 2,000,000	Cycles 100,000	Cycles 2,000,000
Fatigue strengths of flat-plate specimens with and without butt welds, as-welded and with reinforcement removed flush, psi x 10^{-3}				
Plain plate (A-7 steel)	47.8	31.7	26.8	17.5
Transverse butt-welded joint (single V) as welded	34.9	23.2	22.4	14.9
Transverse butt-welded joint (a) single-V; (b) single-U - reinforcement off	37.5(a)	28.7(a)	27.4(b)	16.0(b)
Longitudinal butt-welded joint (single-V) — as welded	39.6	26.0	-	-
Longitudinal butt-welded joint (a) single-V; (b) double V - reinforcement off	47.1(a)	30.5(a)	21.0(b)	15.6(b)
Fatigue strengths of fillet welds of various arrangements in specimens designed for failure in the welds)				
Fillet-welded lap joints - 5/16 in. transverse welds. Failure in welds	30.3	18.5+	16.2	11.3
Fillet-welded lap joints - 5/16 in. longitudinal welds. Failure in welds	27.2	19.7	15.2	10.7
Fillet-welded lap joints - 5/16 in. transverse and longitudinal welds. Failure in welds.	28.3	20.5	13.1	8.9
Tee joints - 5/16 in. fillet welds. Failure in welds.	19.1	9.6	13.3	6.2
Moment connectors - 5/16 in. vertical fillet welds. Failure in welds	46.8	26.5	26.6	17.3
Fatigue strengths of rolled beams with lateral connection plates attached to the tension flanges				
Beams with welded attachments. Longitudinal welds	24.4	13.6	-	-
Beams with welded attachments. Transverse welds	19.2	12.2	-	-
Fatigue Strengths of Welded Beams and Girders				
Rolled beams, 12-in., 31.8 lb I, with continuously welded, partial length cover plates, 4 x 9/16 in. and 3/8 in. transverse fillet welds across square ends of cover plates	22.8	12.0	-	-
Rolled beams, 12-in., 31.8 lb I, with continuously welded, full-length cover plates 4 x 9/16-in. top plate and 6 x 3/8 in. bottom plate, 3/16-in. fillet welds	41.1	22.8	-	-
Built-up girders with continuous web-to-flange fillet welds and with middle stiffeners welded only to compression flange and upper half of web	46.7	17.6	-	-
Fatigue Strengths of Plug- and Slot-Welded Joints (Stresses given are those in welds)				
Plug welds - weld failure	23.1	12.6	11.3	6.5
Slot welds - transverse (balanced design for weld and plate failure)	20.0	10.2	10.0	5.3
Slot welds - longitudinal (balanced design for weld and plate failure)	27.9	11.1	16.1	6.1
Fatigue Strength of Plug-Welded Lap Joint Designed to Fail in Plate Material (Stresses given are those in Plate Material)				
Plug-welded lap joint - plate failure	26.3 24.1	10.1 11.7	14.2 14.1	5.3 6.6

TABLE 7-26 TENSILE STRENGTH OF MIG AND TIG BUTT JOINTS

Alloy	5086	5456	6061-T6
Filler Alloy	5356	5556	5356
Average Ult Tensile, psi	39,000	46,000	30,000
Min Annealed Ult Tensile, psi	35,000	42,000	24,000
Average Yield, psi	19,000	23,000	17,000
Min Yield, psi	17,000	20,000	18,000
Free Bend Elongation, %	38	28	25
Tensile Elongation in 2 in., %	17	14	10

TABLE 7-27 SHEAR STRENGTH OF WELDED ALUMINUM ALLOY

Alloy	*5356*	*5556*
Typical Longitudinal Shear, psi	21,000	26,000
Min Longitudinal Shear, psi	17,000	19,500
Typical Transverse Shear, psi	34,000	41,000
Min Transverse Shear, psi (double fillet)	26,000	30,000

a. Applications
b. Surface preparation
c. Adhesive classifications
d. Design of an adhesive-bonded joint
e. Adhesive selection
f. Safety precautions

This method of fastening aluminum is advancing rapidly with new products and techniques being introduced frequently. Adhesive bonding is being used for the foils on hydrofoil craft, helicopter rotor blades, and other high performance components. Table 7-28 compares the properties of epoxy adhesives. It is recommended that the manufacturers of adhesives be consulted for the latest information before designing adhesive-bonded aluminum structures.

7-3.4.3 Miscellaneous Details

The following list includes many items previously covered. They are listed here to prevent their neglect in the vast amount of calculations and specifications:

a. Areas most frequently damaged include:

(1) Quarters where the sides and transom intersect

(2) Cargo deck area, including all adjacent vertical surfaces

(3) Cab accessories and the tires when the vehicle rolls when moored alongside a ship. The tires can be used effectively as fenders if they are adequately supported.

b. Adequate hand and foot holds must be provided. These may be used to secure gear if a line can be tied to them.

c. Drain plugs should be easily accessible so they can be replaced (with water in the bilge). Plugs with a "T" handle and a guide will prevent their loss and assist in securing in an emergency. The material used for the plug must be compatible with the hull material. An aluminum plug in an aluminum hull will seize.

d. Vibration will be present in any prototype. Addition of some extra panel stiffeners or braces should be anticipated to reduce the amplitude for any new design.

e. Piping penetrating the hull should have a

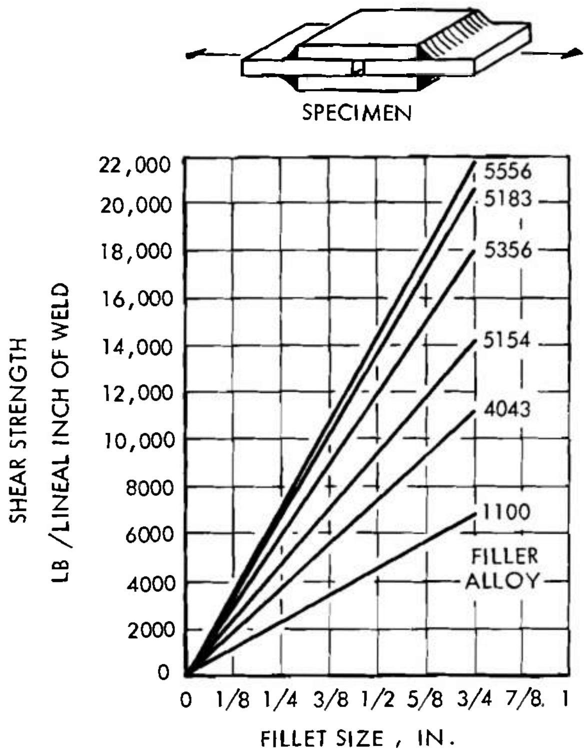


Figure 7-43. Typical Transverse Shear Strengths of Aluminum Fillet Welds

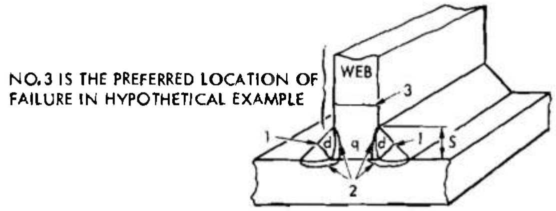


Figure 7-44. Three Possible Areas of Failure in a Double Fillet Weld

flexible section so that it will not vibrate with respect to the hull.

f. Good drainage is necessary for minimum maintenance.

g. Hatches for the removal of all machinery components should be provided. It is recommended that the cargo deck be solid. Hard points to attach lifting gear should be provided in the design stage. Foundation extensions which will allow sliding heavy pieces of machinery to a hatch may be useful.

h. Access inside the hull for maintenance or inspection is usually poor in some locations. Good maintenance will not be maintained with poor access. Likewise maintenance that must be conducted underneath the hull, while it is still dripping, is likely to be frequently overlooked.

i. Ventilation is usually deficient for maintenance work in the hull in hot weather.

Refs. 45 to 49 may be studied to determine past practice with regard to amphibian structural design.

SECTION II PROPULSION SYSTEM DESIGN

7-4 HORSEPOWER REQUIREMENTS

In the design of a propulsion system for a wheeled amphibian, the first step should be a determination of the required horsepower. In general the power requirements will be determined by the mission requirements or military characteristics defined in a QMR or SDR. It is the task of the amphibian designer to establish the minimum power required to satisfy the performance requirements given in the QMR. In some cases the amphibian designer may be faced with the converse problem in which the power plant is specified and the performance must be estimated. In either case the same basic data and design procedures apply.

Among military vehicles the amphibian is unusual in that it has performance requirements on or in three media, viz., water, land, and the interfaces between water and land. Performance in all of these media must be checked to determine which will determine the required power. The paragraphs of this section, which follow, present design theories and data for use in defining the power requirements.

7-4.1 WATERBORNE POWER REQUIREMENTS

In the LARC series of amphibians and the existing high speed amphibian prototypes, the power required is determined by the demands for waterborne performance. In future wheeled amphibians, with the current trend toward improved waterborne performance, this will almost certainly be the case. The waterborne

TABLE 7-28 COMPARISON OF PROPERTIES FOR EPOXY ADHESIVES(1)

Type	*Tensile Shear Strength*,(2) psi	*T-Peel*(3) *Strength*, lb/in.
1. One-component (Heat cured)		
a. High-strength Flexible	4,000-7,000	12-160
b. High-strength Semirigid	4,500-5,500	2-6
2. Two component (Room-temperature-cured)	1,000-4,000	2-30

(1) All test specimens were chemically cleaned—chromic acid-sulfuric acid solution.
(2) Tensile-shear strength—Alclad 2024-T3 1/2 in. lap.
(3) Pull in lb/in. at a right angle to a base plate necessary to peel 0.009 in. thick 5052-H18 alloy from a rigid plate.

performance requirement is generally expressed in terms of a speed. In the case of current displacement hull amphibians (the LARC's), the required performance is expressed in terms of a top speed in calm, deep water. In the case of high speed amphibians, the required performance is expressed in terms of a sustained speed in a given sea state. Other waterborne performance criteria which may affect the power required include navigation in rough water, surfing ability, bollard pull, reversing, and maneuvering. The applicable performance requirement will be defined in the QMR.

The basic relationship between the power delivered to the propulsor and the amphibian's speed, resistance or drag, and propulsor efficiency is given by

$$DHP = \frac{EHP}{\eta_p} = \frac{R_T V}{550\,\eta_p} \tag{7-54}$$

where

DHP = power delivered to the propulsors, hp
EHP = effective horsepower, hp
R_T = total resistance, lb
V = vehicle speed, ft/sec
η_p = propulsive efficiency, nd (par. 7-7.4.3.4)

To determine the delivered power it is necessary to obtain both the total resistance and the propulsive efficiency. Par. 7-7 deals with the selection and design of the propulsor and the calculation of the propulsive efficiency. The remainder of this section deals with the total resistance term of Eq. 7-54. For information on power requirements as determined by bollard pull, reversing, and maneuvering; reference should be made to pars. 7-7 and 7-9. Chapter 9 presents data on the resistance of high-speed amphibians.

7-4.1.1 Elements of Amphibian Waterborne Resistance

The waterborne resistance of an amphibian or marine vehicle can be considered as being made up of the following four main components:

1. The frictional resistance due to the motion of the hull through a viscous fluid
2. The eddymaking resistance due to the energy carried away in eddies shed from the hull, wheels, and appendages
3. The wavemaking resistance due to energy which must be supplied by the craft to the wave system created on the water surface
4. The aerodynamic resistance due to the passage of the above-water part of the craft through the air

Of these resistance components, Components 2 and 3 are of greatest importance to an amphibian and 4 is of little importance. It is common to consider Components 2 and 3 of the resistance together under the term "residuary resistance".

In general the flow conditions on the surface around an amphibian or other marine vehicle are much too complex to allow a theoretical calculation of resistance which is useful in design

work. As a result, experimental data must be used. In order to make the greatest possible use of these data, the principles of dimensional analysis are employed. Dimensional analysis is based on the principle that every equation which expresses a physical relationship must be dimensionally homogeneous. The three basic dimensions in mechanics are mass, length, and time (M, L, and T) from which the dimensions of other quantities—such as acceleration, force, and pressure—can be derived. To apply dimensional analysis to a problem, it is only necessary to know the variables which govern the results. Dimensional solutions do not provide numerical answers themselves but provide the form of the answers. As a result, experimental data may be interpreted in a form most useful in obtaining an empirical solution. Refs. 4 and 53 present the theory and procedures of dimensional analysis in detail.

In applying dimensional analysis to the resistance of a waterborne vehicle, the following variables are assumed to govern the relationship between two geometrically similar bodies (geosims) of different size.

a. Speed V, ft/sec

b. Size of body, as represented by the linear dimension L, ft

c. Mass density of the fluid ρ, slug/ft^3

d. Viscosity of fluid μ, slug/ft-sec

e. Acceleration due to gravity g, ft/sec^2

f. Pressure per unit area in the fluid p, lb/ft^2

Dimensional analysis then indicates that the form of the relationship between the resistance R and the above variables is given by

$$R \propto (\rho V^2 L^2) f\left[\frac{\rho VL}{\mu}, \frac{gL}{V^2}, \frac{p}{\rho V^2}\right] \quad (7\text{-}55)$$

where f is a symbol indicating "some function of". The kinematic viscosity ν (the ratio μ/ρ) and the wetted surface S or other area which is proportional to L^2 are commonly used in Eq. 7-55. This equation is then written in the form

$$\frac{R}{\frac{1}{2}(\rho V^2 S)} = f\left[\frac{VL}{\nu}, \frac{V^2}{gL}, \frac{p}{\rho V^2}\right] \quad (7\text{-}56)$$

The ½ is introduced into the left hand side of Eq. 7-56 to make the pressure (ρV^2) numerically identical with the dynamic pressure in a fluid as defined by Bernoulli's equation. Experimental data must be used to determine the form of the relationship between resistance and the nondimensional parameters VL/ν, $V^2/(gL)$ and $p/(\rho V^2)$.

William Froude in England in 1868 first proposed the form of the relationship between the resistance and the above parameters. His basic concept is still in use. He assumed that the total resistance can be divided into two components, i.e., frictional resistance and residuary resistance. The frictional resistance is assumed to be a function of the parameter VL/ν only and the residuary resistance as a function only of $V^2/(gL)$. The parameter $p/(\rho V^2)$ is a form of the Cavitation Number discussed in par. 7-7.4.4.1 and is assumed not to affect the resistance components except at very high speeds on specialized craft.

7-4.1.1.1 Frictional Resistance

The frictional resistance due to the passage of the hull and appendages through a viscous fluid typically amounts to 50 to 80 percent of the total resistance for a conventional ship or boat form and considerably less for amphibians. Following the concepts presented in par. 7-4.1.1, a frictional resistance coefficient can be defined as

$$C_F = \frac{R_F}{\frac{1}{2}(\rho V^2 S)} \quad (7\text{-}57)$$

where

C_F = frictional resistance coefficient, nd
R_F = frictional resistance, lb
S = surface area, ft^2

and other terms are as previously defined.

For geosims, C_F is only a function of the parameter VL/ν. This follows from the work of Osborne Reynolds, published in 1893, for which reason this parameter is known as the Reynolds number Re. For convenience in calculations of C_F and Reynolds number, Tables 7-29 and 7-30 from Ref. 14 which give the mass density and kinematic viscosity of fresh and salt water are included here.

For the purpose of making ship resistance estimates, considerable effort has been devoted to determining the relationship between the frictional resistance coefficient C_F and the Reynolds number Re. The development of several proposed relationships for smooth flat plates is covered in Ref. 14. Three of these relationships are given in Fig. 7-45. These

TABLE 7-29 VALUES OF MASS DENSITY ρ FOR FRESH AND SALT WATER

Values as adopted by the ITTC meeting in London, 1963.
Salinity of salt water 3.5 percent.

Density of fresh water ρ, slug/ft^3	*Temp*, °F	*Density of salt water ρ*, slug/ft^3	*Density of fresh water ρ*, slug/ft^3	*Temp*, °F	*Density of salt water ρ*, slug/ft^3
1.9399	32	1.9947	1.9384	59	1.9905
1.9399	33	1.9946	1.9383	60	1.9903
1.9400	34	1.9946	1.9381	61	1.9901
1.9401	35	1.9945	1.9379	62	1.9898
1.9401	36	1.9944	1.9377	63	1.9895
1.9401	37	1.9943	1.9375	64	1.9893
1.9401	38	1.9942	1.9373	65	1.9890
1.9401	39	1.9941	1.9371	66	1.9888
1.9401	40	1.9940	1.9369	67	1.9885
1.9401	41	1.9939	1.9367	68	1.9882
1.9401	42	1.9937	1.9365	69	1.9879
1.9401	43	1.9936	1.9362	70	1.9876
1.9400	44	1.9934	1.9360	71	1.9873
1.9400	45	1.9933	1.9358	72	1.9870
1.9399	46	1.9931	1.9355	73	1.9867
1.9398	47	1.9930	1.9352	74	1.9864
1.9398	48	1.9928	1.9350	75	1.9861
1.9397	49	1.9926	1.9347	76	1.9858
1.9396	50	1.9924	1.9344	77	1.9854
1.9395	51	1.9923	1.9342	78	1.9851
1.9394	52	1.9921	1.9339	79	1.9848
1.9393	53	1.9919	1.9336	80	1.9844
1.9392	54	1.9917	1.9333	81	1.9841
1.9390	55	1.9914	1.9330	82	1.9837
1.9389	56	1.9912	1.9327	83	1.9834
1.9387	57	1.9910	1.9324	84	1.9830
1.9386	58	1.9908	1.9321	85	1.9827
			1.9317	86	1.9823

Note: For other salinities, interpolate linearly.

friction lines are based on turbulent flow.

In practice, the hull of a marine vehicle is neither flat nor smooth. As a result, in estimating the frictional resistance some increase is made in the values of C_F given in Fig. 7-45. For standard ship resistance estimates, it is customary to increase the value of C_F by 0.0004. The ATTC line +0.0004 roughness, or correlation allowance ΔC_F, is also shown in Fig. 7-45. In the event that it is necessary to make a resistance estimate for an amphibian form in which the effect of Reynolds number on C_F is important (see par. 7-4.1.1.3) it would be reasonable to use a roughness allowance of 0.0004.

7-4.1.1.2 Residuary Resistance

Those components of the total resistance due to eddymaking and wavemaking are commonly

TABLE 7-30 VALUES OF KINEMATIC VISCOSITY ν FOR FRESH AND SALT WATER

Values adopted by the ITTC meeting in London, 1963.
Salinity of salt water, 3.5 percent.

Kinematic viscosity of fresh water ν, $\frac{ft^2}{sec}$	*Temp,* °F	*Kinematic viscosity of salt water* ν, $\frac{ft^2}{sec}$	*Kinematic viscosity of fresh water* ν, $\frac{ft^2}{sec}$	*Temp,* °F	*Kinematic viscosity of salt water* ν, $\frac{ft^2}{sec}$
1.9231×10^{-5}	32	1.9681×10^{-5}	1.2260×10^{-5}	59	1.2791×10^{-5}
1.8871	33	1.9323	1.2083	60	1.2615
1.8520	34	1.8974	1.1910	61	1.2443
1.8180	35	1.8637	1.1741	62	1.2275
1.7849	36	1.8309	1.1576	63	1.2111
1.7527	37	1.7991	1.1415	64	1.1951
1.7215	38	1.7682	1.1257	65	1.1794
1.6911	39	1.7382	1.1103	66	1.1640
1.6616	40	1.7091	1.0952	67	1.1489
1.6329	41	1.6807	1.0804	68	1.1342
1.6049	42	1.6532	1.0660	69	1.1198
1.5777	43	1.6263	1.0519	70	1.1057
1.5512	44	1.6002	1.0381	71	1.0918
1.5254	45	1.5748	1.0245	72	1.0783
1.5003	46	1.5501	1.0113	73	1.0650
1.4759	47	1.5259	0.9984	74	1.0520
1.4520	48	1.5024	0.9857	75	1.0392
1.4288	49	1.4796	0.9733	76	1.0267
1.4062	50	1.4572	0.9611	77	1.0145
1.3841	51	1.4354	0.9452	78	1.0025
1.3626	52	1.4142	0.9375	79	0.9907
1.3416	53	1.3935	0.9261	80	0.9791
1.3212	54	1.3732	0.9149	81	0.9678
1.3012	55	1.3535	0.9039	82	0.9567
1.2817	56	1.3343	0.8931	83	0.9457
1.2627	57	1.3154	0.8826	84	0.9350
1.2441	58	1.2970	0.8722	85	0.9245

Note: For other salinities, interpolate linearly.

known jointly as the residuary resistance. A residuary resistance coefficient can be defined as

$$C_R = \frac{R_R}{\frac{1}{2}(\rho V^2 S)} \tag{7-58}$$

where

S = surface area, ft^2
C_R = residuary resistance coefficient, nd
R_R = residuary resistance, lb

It has been shown that for the purpose of engineering estimates C_R can be considered to be a function of the parameter $V^2/(gL)$ so that geosims operating at the same value of $V^2/(gL)$ have the same value of C_R. Since Froude first suggested this, the parameter $V^2/(gL)$ is known as the Froude number F_N. The Froude number often is given in references in the form $V/\sqrt{gL}$ which is also nondimensional. A common

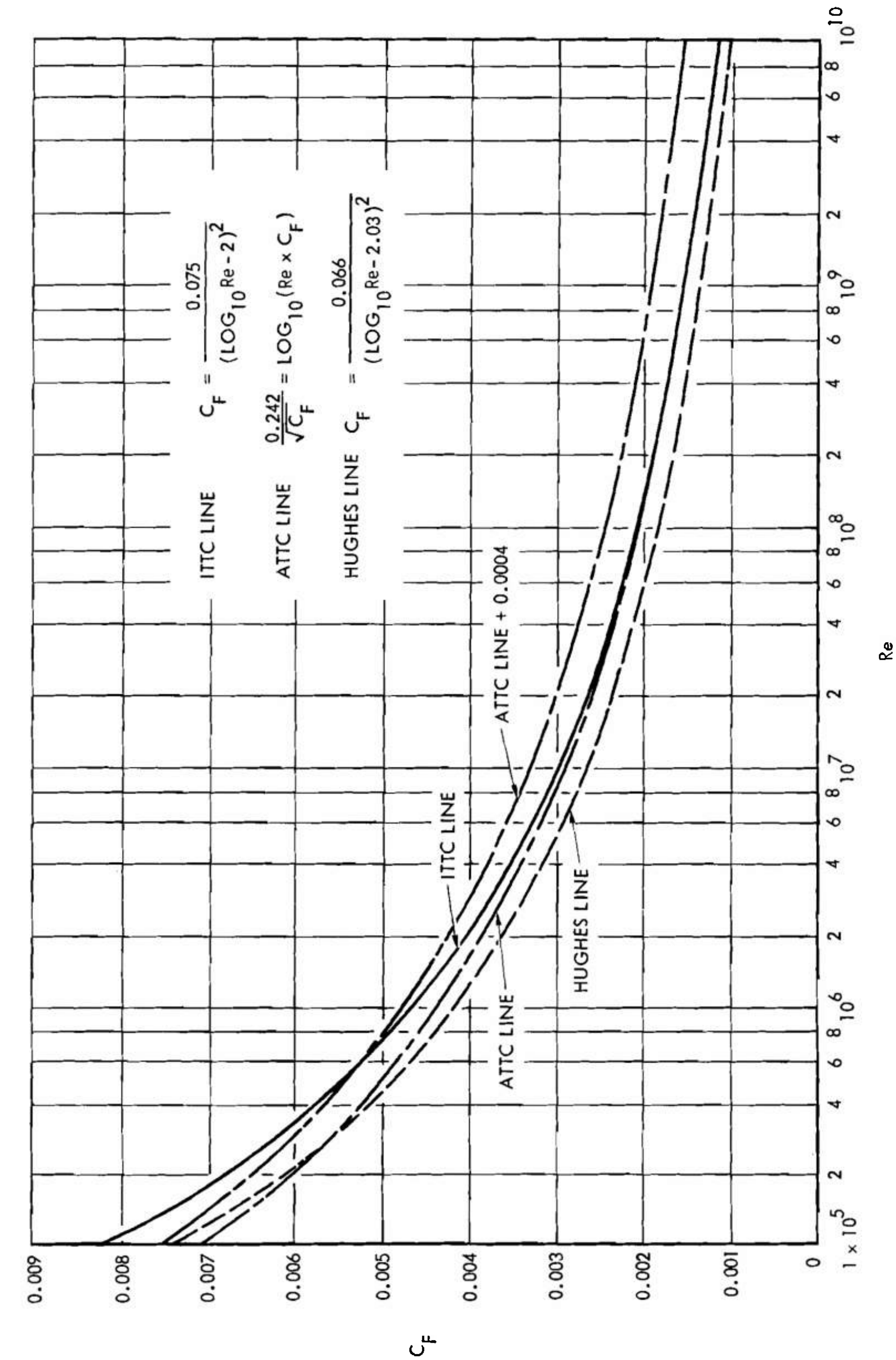


Figure 7-45. Standard Skin Friction Lines

dimensional form of the Froude number is the speed-length ratio $V_K/\sqrt{L}$. When V_K is given in knots, L in ft, and g in ft/sec^2; the relationship between the two is

$$\frac{V}{\sqrt{gL}} = F_N = 0.298\frac{V_K}{\sqrt{L}}$$

or

$$\frac{V_K}{\sqrt{L}} = 3.355\,F_N \tag{7-59A}$$

It is important to note that geosims operating at the same Froude number or speed-length ratio will have the same ratio of residuary resistance to displacement, regardless of size. In fact, if the effects of Reynolds number on C_F are neglected, geosims at the same Froude number will have the same ratio of total resistance to displacement, regardless of size. The proof of this is given in par. 7-4.1.1.3.

7-4.1.1.3 Scaling Laws and Model Tests

In the design of an amphibian the prediction of resistance must depend on data from model tests. The expansion of the measured resistance of a model to the resistance of the full scale craft is based on the relationship between the components of the resistance, and the Reynolds number and Froude number presented above. The residuary resistance coefficient C_R will be the same for a model and full size craft if they are operating at the same Froude number. Then the ratio of full size resistance to model resistance is given by

$$\frac{R_{Rs}}{R_{Rm}} = \frac{\tfrac{1}{2}\rho_s V_s^2 S_s C_{Rs}}{\tfrac{1}{2}\rho_m V_m^2 S_m C_{Rm}} \tag{7-59B}$$

where the subscripts s and m refer to full size and model, respectively. At the same Froude number C_{Rs} equals C_{Rm} and the ratio of V_s to V_m is given by

$$\frac{V_s^2}{V_m^2} = \frac{L_s}{L_m} \tag{7-60}$$

If Eq. 7-60 is substituted into Eq. 7-59B and ρ_s is assumed equal to ρ_m, then

$$\frac{R_{Rs}}{R_{Rm}} = \frac{L_s S_s}{L_m S_m} = \frac{L_s^3}{L_m^3} = \frac{\Delta_s}{\Delta_m} \tag{7-61}$$

where Δ = displacement, lb.

Thus, the ratio of residuary resistance to displacement is constant between the full scale craft and the model. The relationship among the linear scale ratio λ, the speeds, and resistance ratio of the full size craft and the model are given by

$$\left.\begin{aligned} \frac{L_s}{L_m} &= \lambda \\ \frac{V_s}{V_m} &= \sqrt{\lambda} \\ \frac{R_{Rs}}{R_{Rm}} &= \lambda^3 \end{aligned}\right\} \tag{7-62}$$

In a typical model test ρ_s does not equal ρ_m. As a result, the model is tested at the displacement that gives the correct scale waterline rather than at a displacement determined by the scale ration λ^3.

The relationship given in Eqs. 7-57 and 7-62 would hold true for the case of frictional resistance if the values of C_{F_s} and C_{F_m} were the same for a test with the speeds scaled by Eq. 7-60. However, the values of C_F are equal only if the Reynolds numbers are equal. For equal Reynolds numbers the relationship between full size and model speed is

$$\frac{V_s}{V_m} = \frac{L_m}{L_s} \tag{7-63}$$

Clearly this relationship cannot be satisfied in the same model test that satisfies the relationship given in Eq. 7-60.

The practical means for dealing with this difficulty is by considering the total resistance R_T to be equal to the sums of the frictional resistance and residuary resistance. Thus

$$\frac{R_T}{\tfrac{1}{2}(\rho V^2 S)} = C_T = C_F + C_R \tag{7-64}$$

The expansion of the model data involves taking the C_T obtained at a given Froude number (speed-length ratio) and subtracting from it the value of C_F at the resulting Reynolds number. The remainder is the value of C_R at the given Froude number. The Reynolds number of the full size craft is calculated at the given Froude

number and the value of C_F for the full size craft obtained. This is then added to the C_R from the model, with a ΔC_F (incremental ΔC_F) for roughness to give the C_T, and thus total resistance, of the full size craft. The values of C_F as a function of Reynolds number are obtained from the friction lines illustrated in Fig. 7-45. The expansion procedure described above is illustrated graphically in Fig. 7-46.

For typical wheeled amphibians such as the LARC series, the DRAKE and DUKW, the residuary resistance is very large compared to the frictional resistance due to the effects of wheels and other appendages. Also, the scale ratio is smaller than is usual for a typical ship model test. As a result, the effect of Reynolds number on the C_F difference between model and full size is small, and the resulting change in C_F is a very small part of C_T. Thus it is common practice to expand the model resistance to full scale by the ratio of the displacements, neglecting the effects of Reynolds number. Whether this simplification should be used in model tests of new amphibian designs will depend on the hull form and appendage geometry involved.

7-4.1.2 Amphibian Resistance Data

From past model tests on wheeled amphibian designs, there are data available on vehicle resistance. These data can serve two purposes in the design of a wheeled amphibian. In the early stages of a design, or in the event a model test is not possible, these data can be used to estimate the waterborne resistance. In this case the procedure is to select the design nearest in characteristics to the new design and use the earlier resistance data in making an estimate. Of course, corrections must be made for displacement, different hull shape, and appendages. The influence of these items on resistance is covered in pars. 7-4.1.2.2 and 7-4.1.2.3.

The other use of available resistance data is for making comparisons between new and older designs. If a new design has unexpectedly greater resistance than those of the past, it is probable

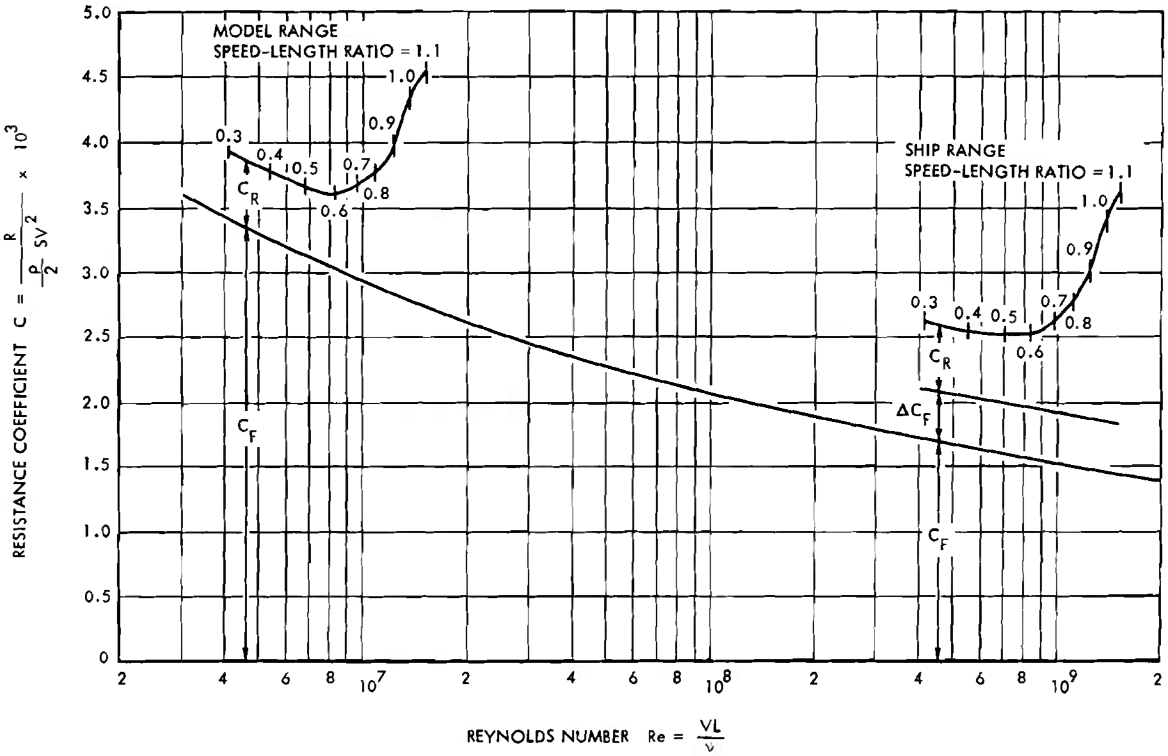


Figure 7-46. Comparison of Resistance Coefficients for Model and Full-scale Vessel

that a redesign effort and further tests would be justified. Refs. 11 and 54 through 60 present model test data on the resistance of past wheeled amphibians.

7-4.1.2.1 Data Plots

The model test data from the references are plotted in a consistent form in Figs. 7-47 through 7-53. These plots are in the form of the total drag-to-displacement ratio as a function of speed-length ratio. Speed is in knots and the length, on the at-rest waterline, in feet. Data are presented for the DUKW, GULL, DRAKE, LARC V, LARC XV, LVW, and LVH-X2. In the case of the LVW and LVH-X2 high-speed amphibians, the data are for operation in the displacement modes.

7-4.1.2.2 Influence of Basic Hull Form on Resistance

In the selection of the basic hull form or in developing a resistance estimate, it is necessary to consider the influence of hull shape. The hull forms used on conventional amphibians are quite different from boat hulls of similar displacement. Amphibians are generally shorter for a given displacement, which results in full hulls with chines at the turn of the bilge and transom. In some vehicles—such as the DUKW, SUPERDUCK, and DRAKE—a scow-type bow has been adopted. The high displacement-length ratio results in high wavemaking drag and the chines cause high eddymaking drag.

The only type of marine craft with displacement-length ratios as large as those of amphibians are landing craft. Fig. 7-54 presents a comparison between the drag-displacement ratio of various landing craft and "faired" hull forms of the DRAKE, LARC V, and LARC XV, as a function of displacement-length ratio and speed-length ratios. Models of the DRAKE, LARC V, and LARC XV were tested with the wheels removed and the wheel pockets filled in to produce a faired hull, thus permitting realistic comparison of the resistance of the basic hull forms without the influence of the wheels. At speed-length ratios around 1.0 the drag-displacement ratios of the "faired" amphibian hull forms are about the same as those of the landing craft. At higher-speed-length ratios the drag-displacement ratios of the amphibian hull forms are higher. This is probably the result of the effects of the propeller tunnels and sloped transoms which prevent the development of dynamic lift as in the case of the landing craft. These features are present in amphibians because of the need for adequate ground clearance and departure angles.

Since it is known that a large part of the total resistance of an amphibian is due to eddymaking, it is useful to compare the total drag coefficients of various amphibian hull forms. The total drag coefficient is similar to the C_F or C_R developed in par. 7-4.1.1 except that it is based on the total drag and the projected frontal area instead of the wetted surface, i.e., $C_D = R_T/(\frac{1}{2}\rho V^2 A)$. When eddymaking drag and frictional drag are the dominant part of the total drag, the total drag coefficient will tend to be constant with speed-length ratio. When wavemaking drag is a large part of the total, the total drag coefficient will increase with speed-length ratio.

The total drag coefficients as a function of speed-length ratio for several amphibians are presented in Fig. 7-55. As expected the "faired" hull forms of the DRAKE, LARC V, and LARC XV have drag coefficients which increase rapidly with speed-length ratio. The LARC's with their ship-like bows have lower drag coefficients than the DRAKE with the scow-type bow. However, it should be noted that for the same projected frontal area the DRAKE has a higher displacement than the LARC's, so that its resistance-displacement ratio is about the same. Thus the choice between the scow form or the ship form bow must not be made on the basis of calm water resistance. Considerations of resistance in waves and seakeeping will also affect the choice between the two.

7-4.1.2.3 Influence of Appendages on Amphibian Resistance

On conventional amphibians, the exposed wheels are the most important appendages. Figs. 7-49 through 7-53 indicate the large increase in resistance between the "faired" hull form and the actual hull with wheels. Depending on the displacement, the presence of wheels and wheel wells increases the resistance 50 to 100 percent.

The effect of appendages on the total drag coefficient is presented in Fig. 7-55. As expected, the effect of the wheels reduces the change in drag coefficient with speed-length

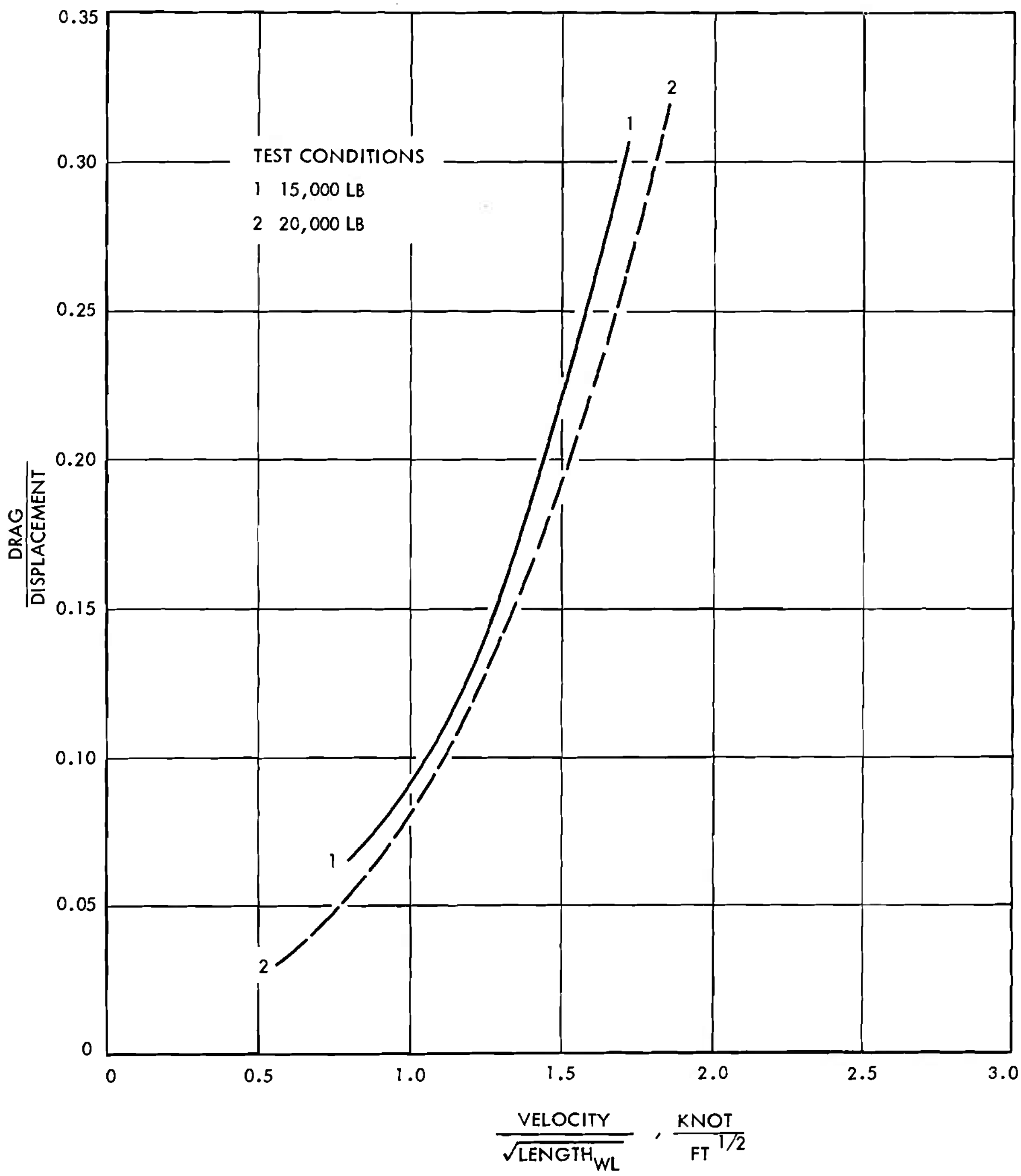


Figure 7-47. Resistance of DUKW

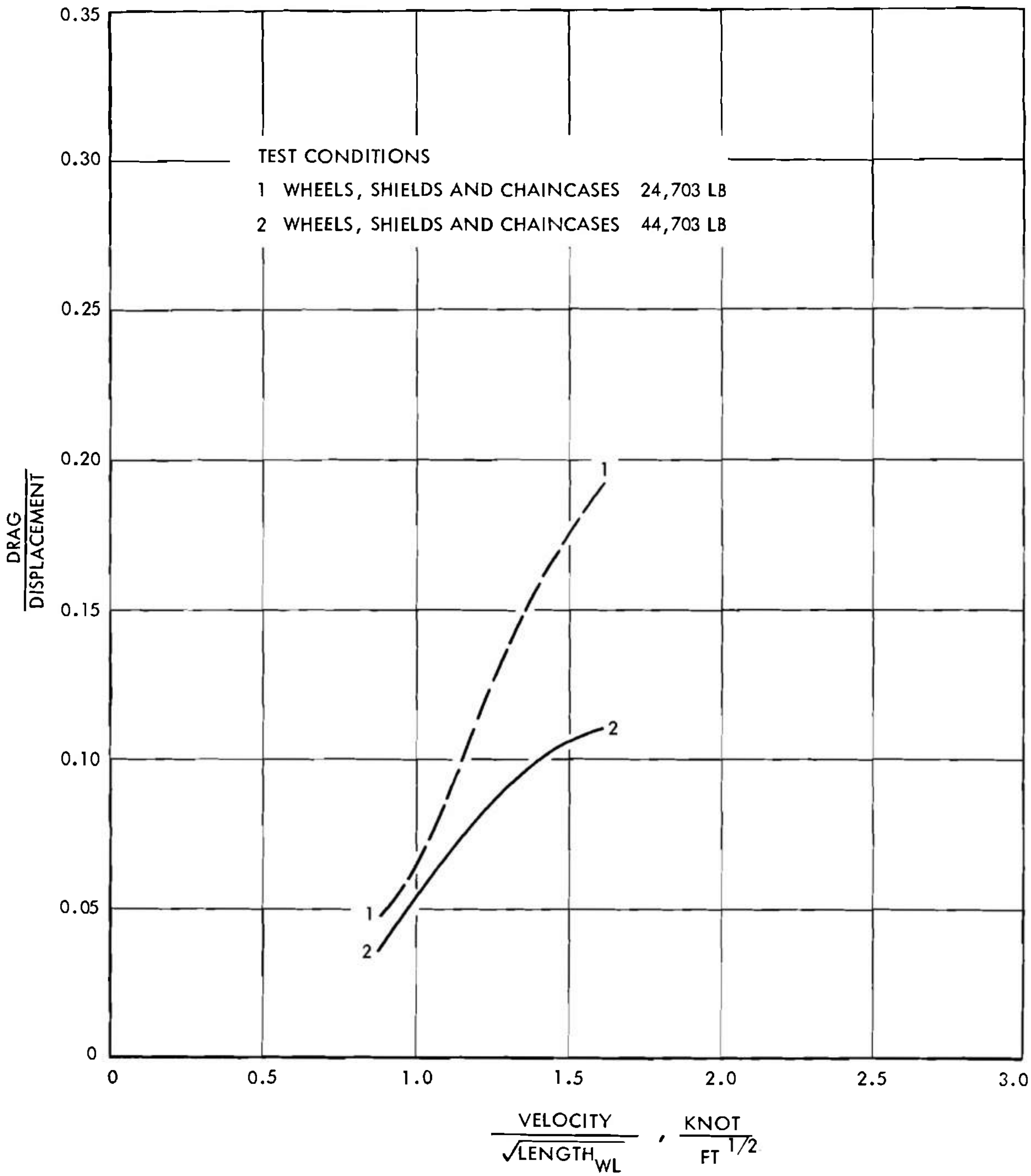


Figure 7-48. Resistance of GULL

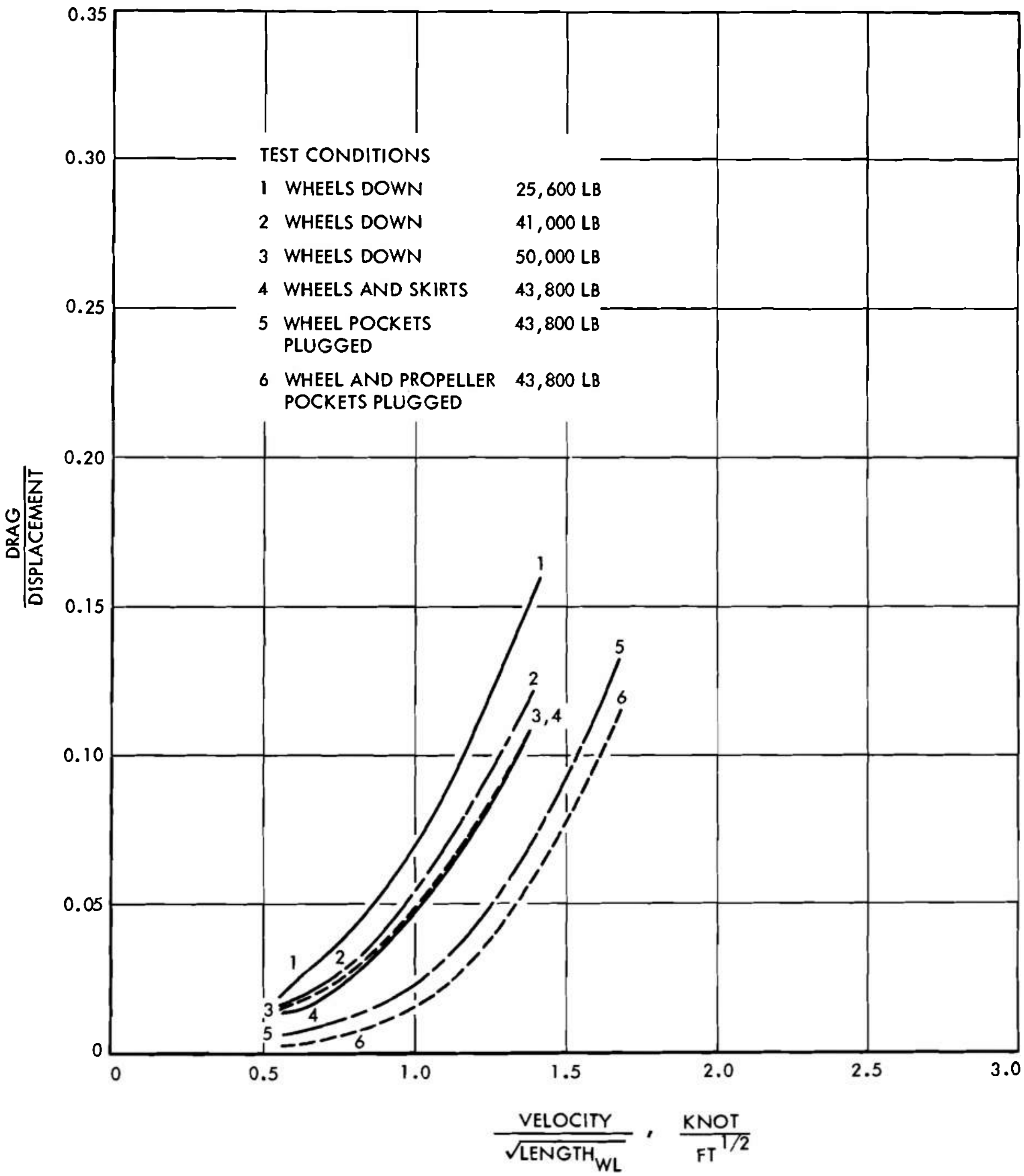


Figure 7-49. Resistance of DRAKE

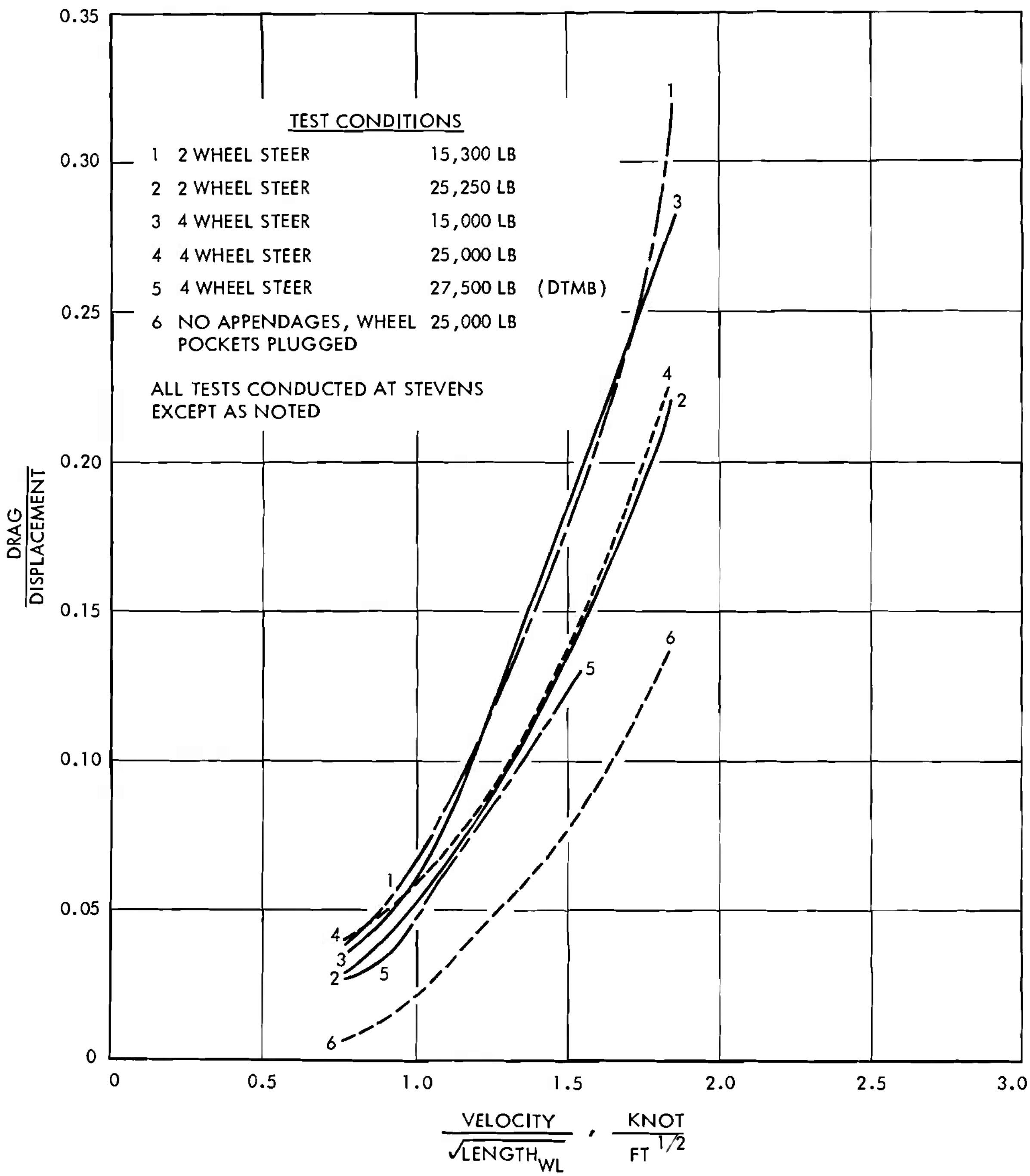


Figure 7-50. Resistance of LARC V

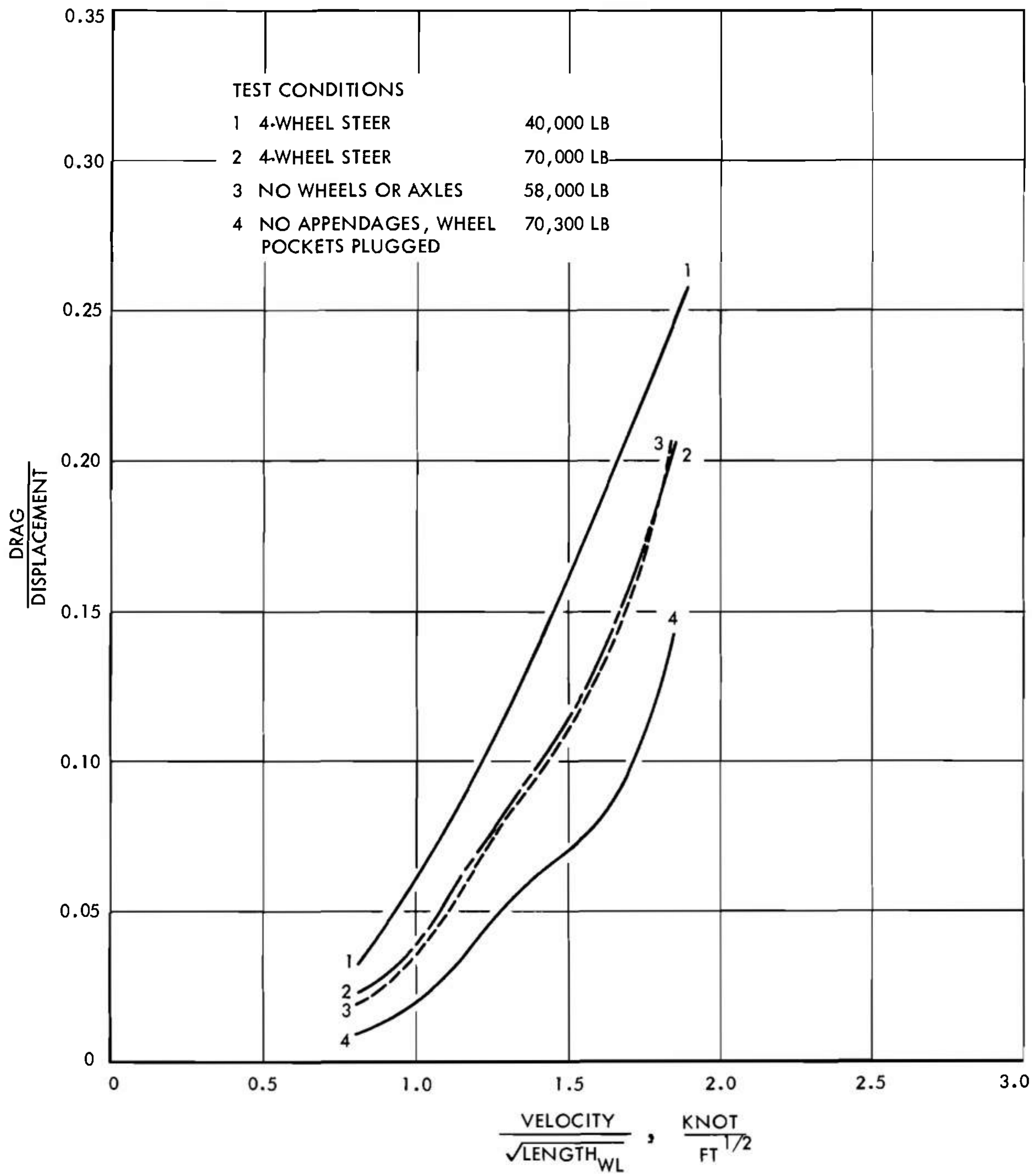


Figure 7-51. Resistance of LARC XV

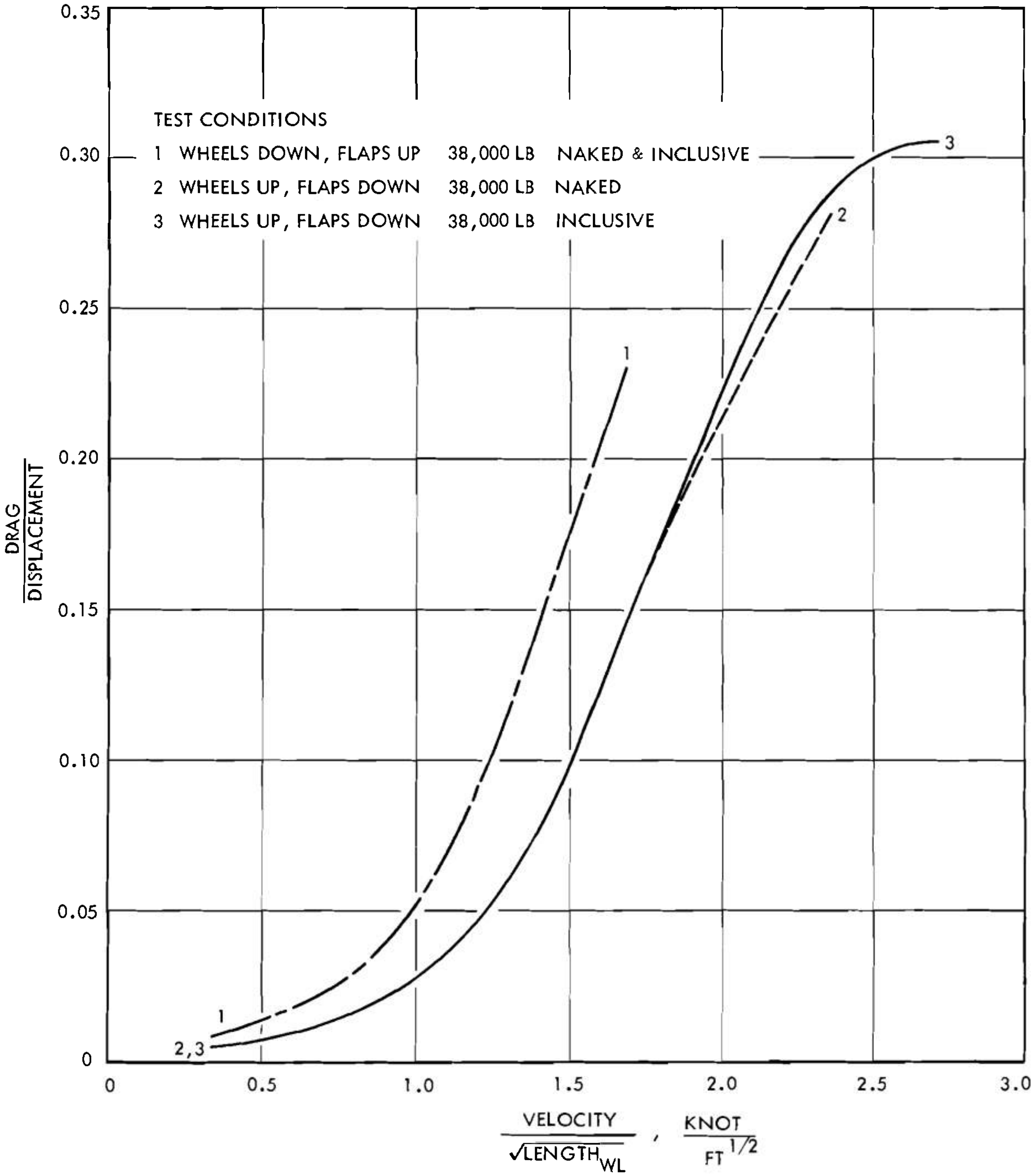


Figure 7-52. Resistance of LVW

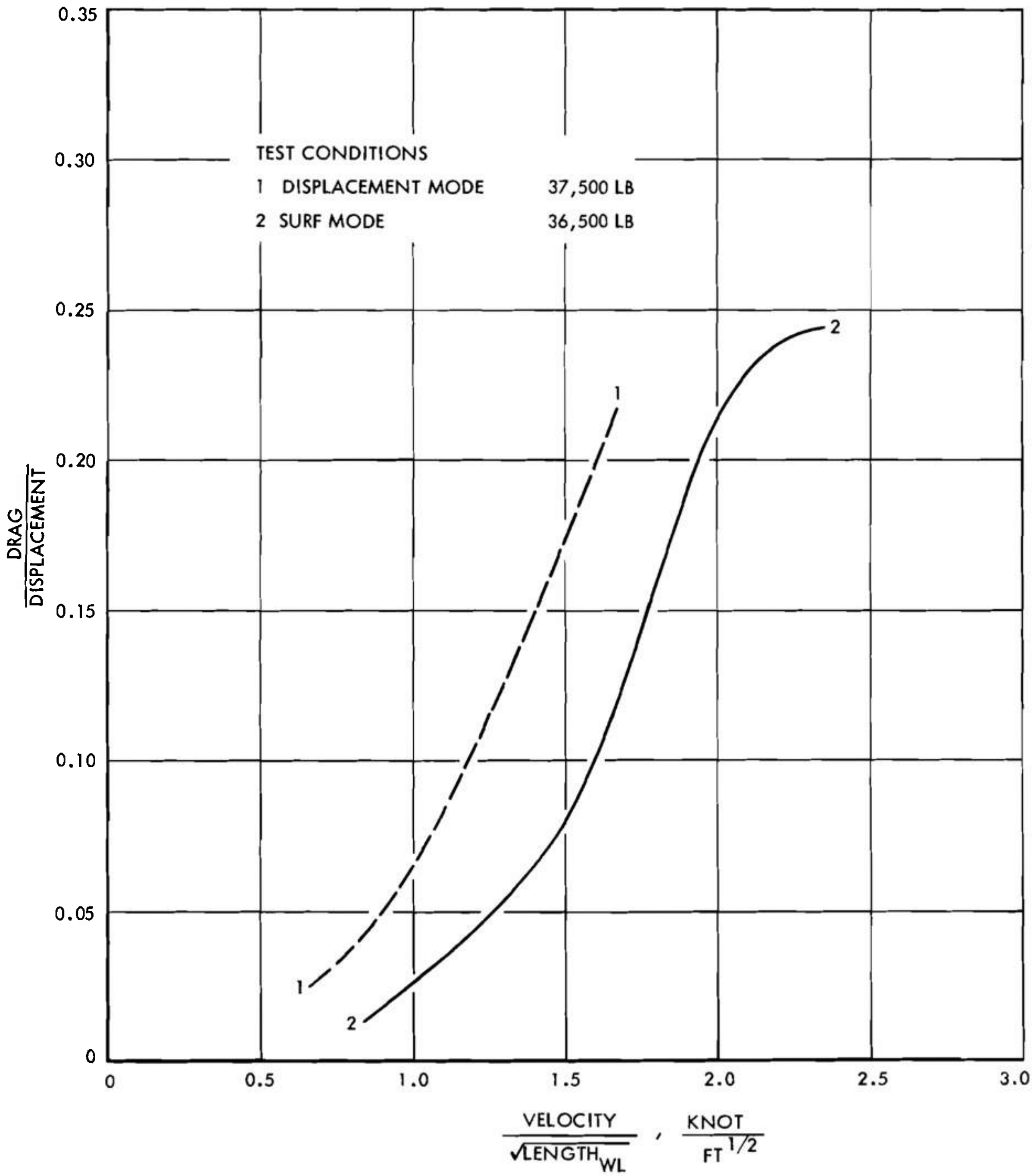


Figure 7-53. Resistance of LVH-X2

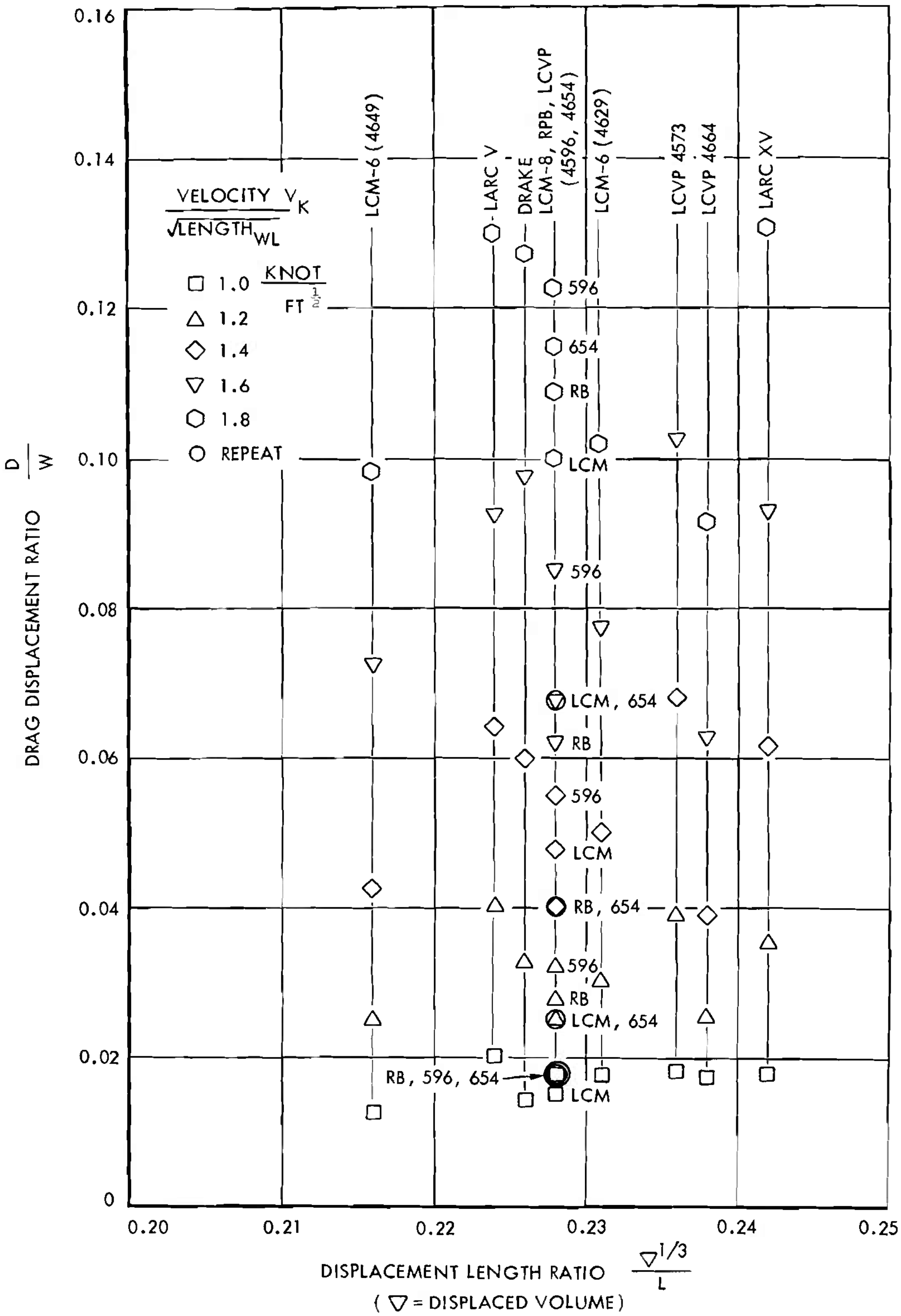


Figure 7-54. Comparison of Amphibian and Landing Craft Resistance

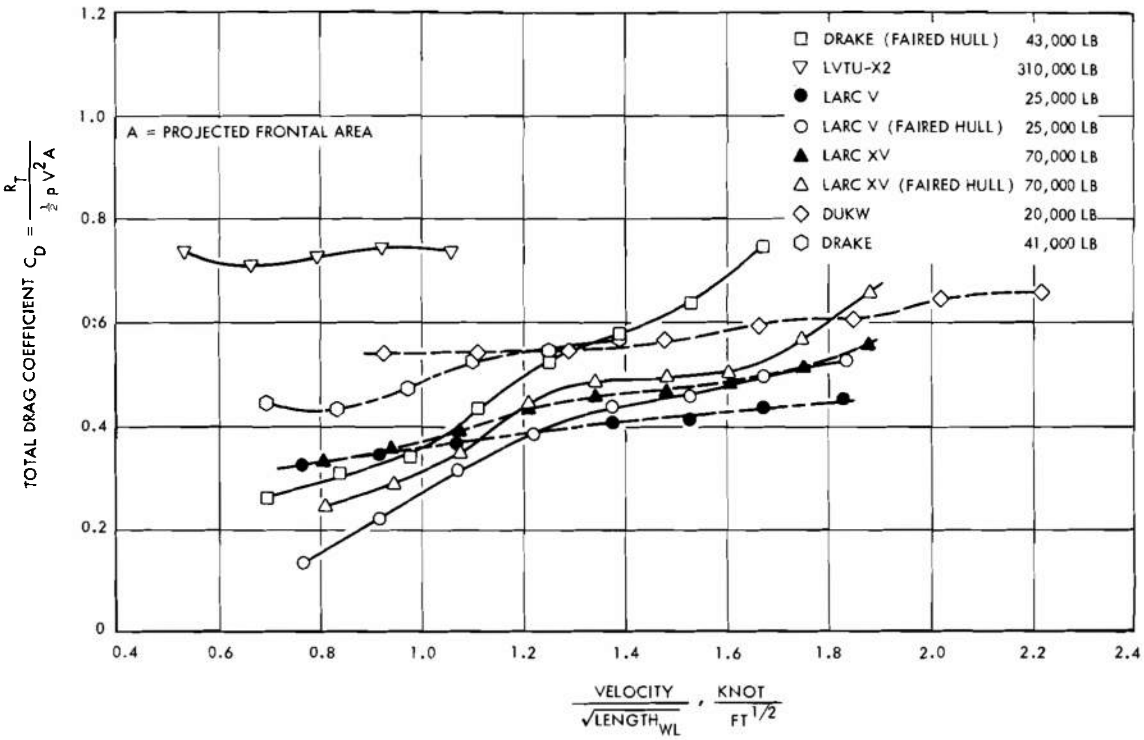


Figure 7-55. Amphibious Vehicle Drag Coefficients

ratio. Since the exposed frontal area of all of the wheels is included in the area term of the drag coefficient, the numerical value of the coefficient does not increase. Data on the drag coefficient of the LVTU-X2 have been included in Fig. 7-55 to indicate the expected upper limits of the drag coefficient of an amphibian with appendages. The LVTU-X2 was a tracked amphibian with an unfaired box type hull form of very high displacement-length ratio.

The apparent drag coefficient of the wheels and wheel wells were obtained for the DRAKE, LARC V, and LARC XV by calculating the difference between the drag of the hull with wheels and the faired hulls. The area used in the drag coefficient is the exposed frontal area of all of the wheels, viz., 4 in the case of the LARC's and 8 for the DRAKE. The resulting drag coefficients are presented in Fig. 7-56. In general, there is a trend of decreasing wheel drag coefficient with speed-length ratio. At speed-length ratios above 1.1 the wheel drag coefficient has a value of 0.42, with all data within = 10 percent of the mean.

7-4.1.2.4 Influence of Rough Water on Amphibian Resistance

Depending on the performance requirements for a given design, it may be necessary to estimate the resistance in waves. As in the case of smooth water resistance, model tests of the design or data from model tests of similar designs must be used in making the estimate. Ideally, a test series should be conducted in a model tank in which the required irregular waves could be generated. Such test facilities do exist (see par. 6-9), but no wheeled amphibian has been tested in these conditions. Model tests of the LARC V and LARC XV were conducted in regular waves (waves of constant height and length) and the results reported in Refs. 11 and 57. In order to make such estimates for an amphibian for which no test data are available, the added resistance from the model tests of the LARC V in head waves has been deduced. These data are presented in Fig. 7-57 as cross plots with respect to speed-length ratio and the number of wave lengths per water line length.

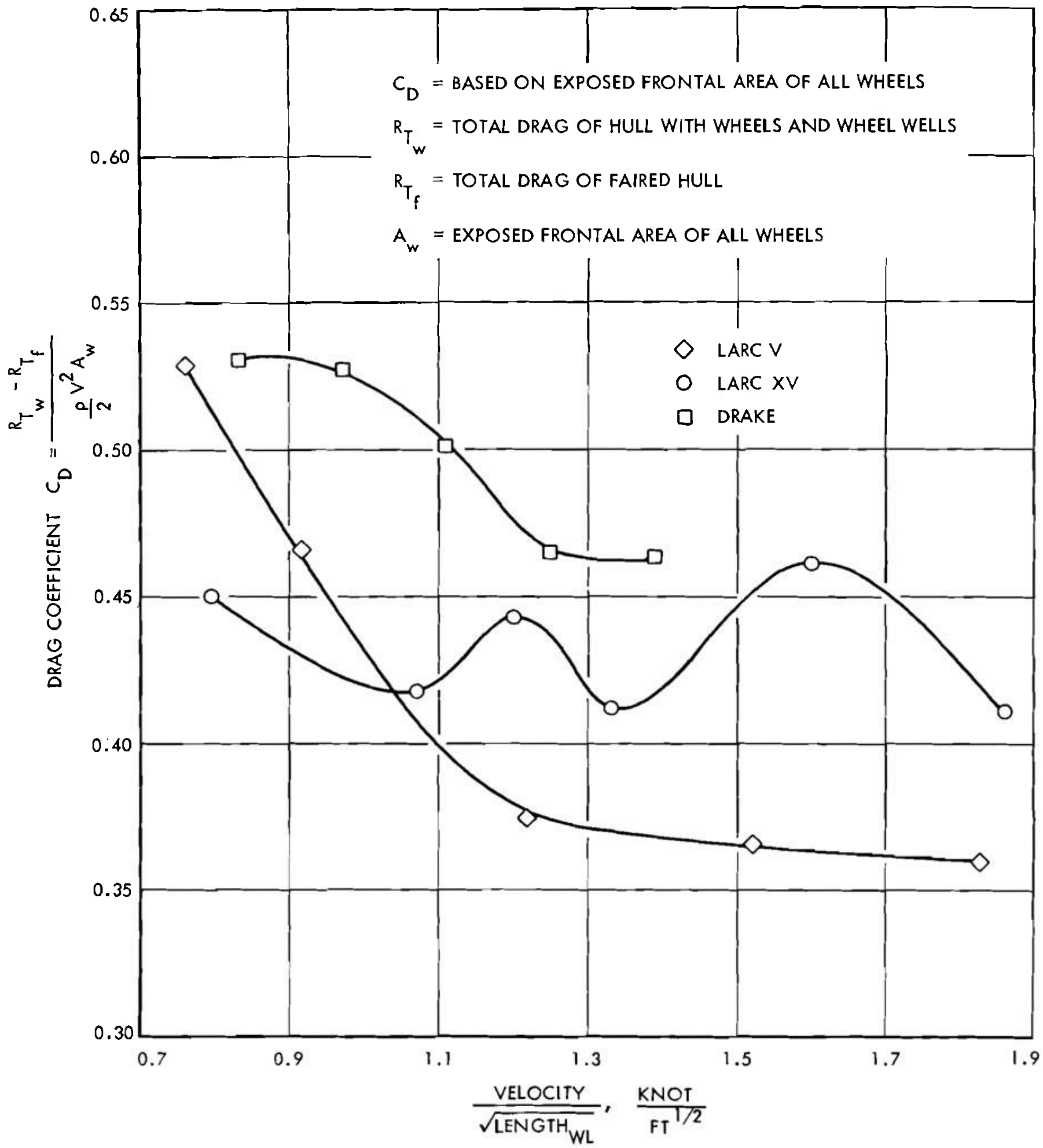


Figure 7-56. Drag Coefficient of Wheels and Wheel Wells

Theory and experiments with conventional ship forms have shown that the resistance increase of a ship in waves is mainly due to the heaving and pitching motions, and the phases of these motions with respect to the waves. The resistance increase is not a function of the smooth water resistance. As a result, the data in Fig. 7-57 should not be sensitive to minor changes in appendages and hull shape.

A procedure for estimating the added resistance in irregular head seas from regular wave data is given in Ref. 61. For theoretically correct estimates, the added resistance must increase as the square of the wave height. Experimental findings on ship forms differ somewhat with theory. However, no other estimating procedure is available at this time. The expression for the average wave-induced added resistance of an amphibian, when proceeding at an average speed in long crested irregular seas from directly ahead, is given by

$$R^*_{AW} = 2\Delta \int_o^\infty \Phi^+_{\xi\xi}(\omega) N_R(\omega) d\omega \qquad (7\text{-}65)$$

$$N_R(\omega) = \frac{R_w}{\Delta A^2}$$

where

R^*_{AW} = average added resistance, lb

Δ = displacement, lb

$\Phi^+_{\xi\xi}$ = wave spectral density ordinate, ft²-sec

R_W = added resistance in regular wave of amplitude A and frequency ω, lb

A = regular wave amplitude, ft

ω = wave frequency, rad/sec

The N_R term of Eq. 7-65 is given as a function of wave length/water line length ratio in Fig. 7-57. The relationship between wave length and frequency for deep water waves is given by

$$\left.\begin{aligned} \lambda &= \frac{64.4\pi}{\omega^2} \\ \omega &= \sqrt{\frac{64.4\pi}{\lambda}} \end{aligned}\right\} \qquad (7\text{-}66)$$

where

λ = wave length, ft

ω = wave frequency, rad/sec

The following example illustrates the calculation procedure, based on Eq. 7-65:

Given: It is required to estimate the average added resistance of the LARC V proceeding in head Sea State 3 at an average speed of 8 mph $V_K/\sqrt{LWL}$ = 1.22 knot/ft$^{1/2}$, where LWL is water line length.

1. It is first necessary to have an analytical description of a Sea State 3. This description must be in the form of a ½ amplitude wave spectrum. This spectrum is a description of the distribution of wave energy as a function of wave frequency. The concept of the wave spectrum is covered in detail in Ref. 62. The wave spectrum in most general use at this time is that due to Pierson and Moskowitz, which is given by:

$$\Phi^+_{\xi\xi}(\omega) = \left(\frac{8.10 \times 10^{-3}\, g^2}{\omega^5}\right) \exp\left[-0.74\left(\frac{g}{U\omega}\right)^4\right] \qquad (7\text{-}67)$$

where

$\Phi^+_{\xi\xi}(\omega)$ = wave spectral density ordinate, ft²-sec

g = acceleration due to gravity, ft/sec²

ω = wave frequency, rad/sec

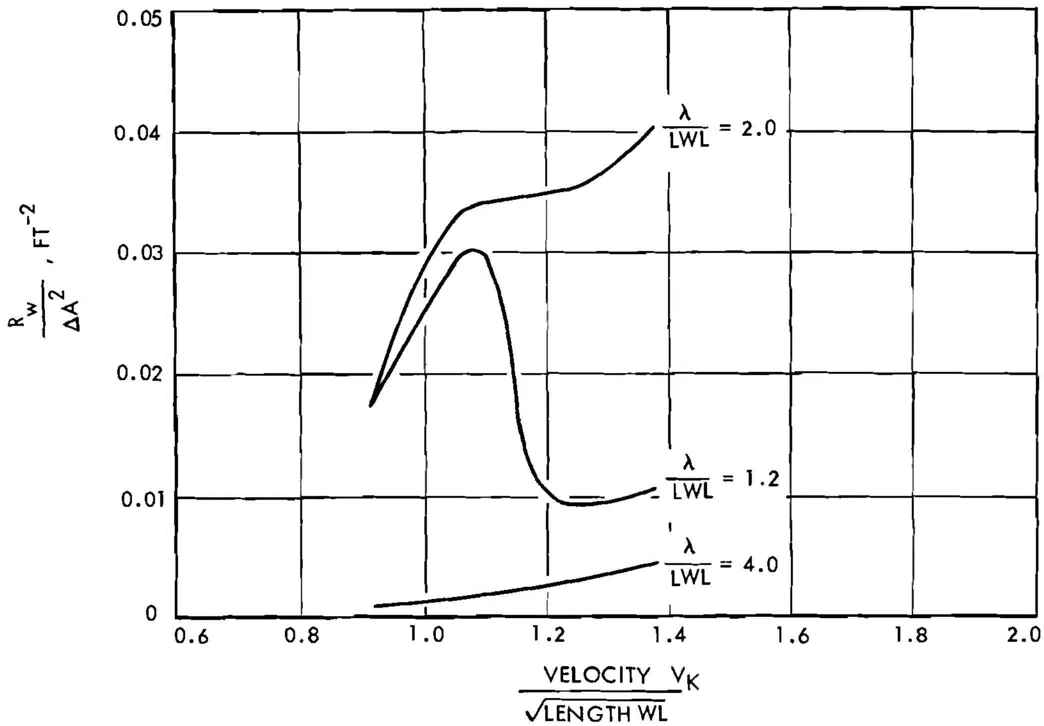
U = wind speed, ft/sec

It is assumed that a Sea State 3 is described by a Pierson-Moskowitz spectrum for a wind speed of 15 knots.

2. The integration in Eq. 7-65 can best be carried out numerically using Simpson's rule since the analytical expression for N_R as a function of ω is not available. Fig. 7-57 indicates that the numerical integration need be carried out only for a range of ω that covers the range of λ/LWL from 1.0 to 4.0. Outside of this range N_R equals zero. The details of the calculation are given in Table 7-31.

7-4.1.3 Design Waterborne Power Estimate

From the previous paragraphs it is possible to define some general considerations that apply to the determination of required waterborne power. If the amphibian design is to be a success, the power required must be determined on the basis of the most demanding quantitative performance requirement in the QMR or other specification. This consideration is of the greatest importance. If at all possible the waterborne power required should be determined from a model test that, as closely as possible, duplicates the conditions given in the QMR or



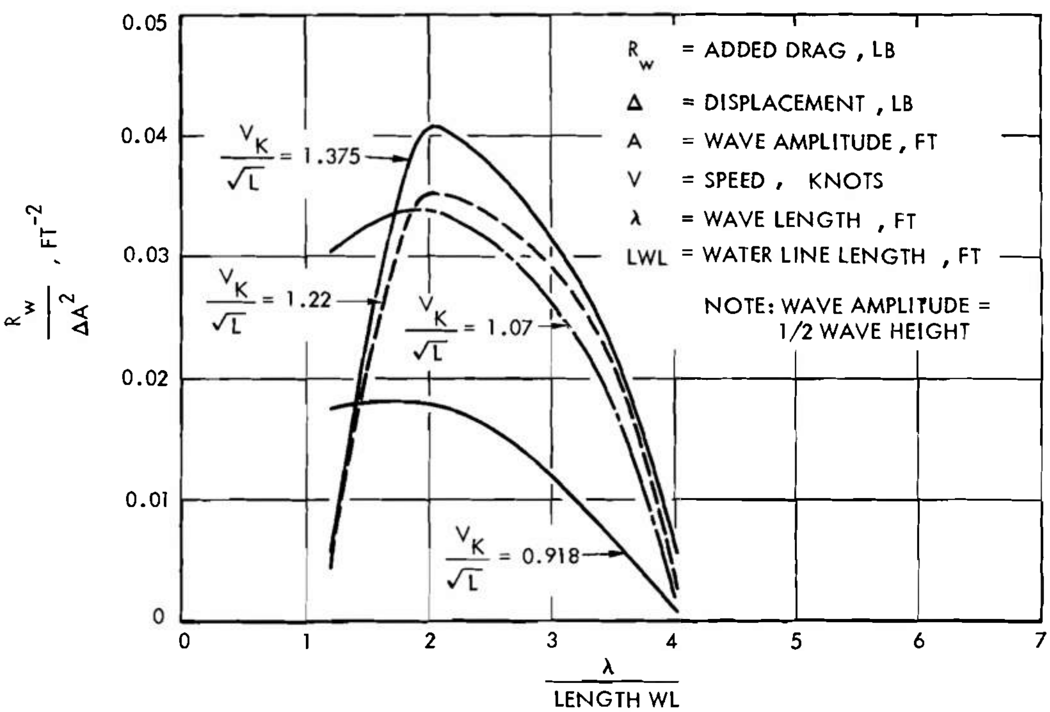


Figure 7-57. LARC V Added Resistance in Regular Head Waves

TABLE 7-31 ESTIMATE OF ADDED RESISTANCE OF LARC V IN SEA STATE 3

Average Speed = 8 mph; $\frac{V_K}{\sqrt{LWL}} = 1.22$ knot/ft$^{1/2}$

(1) ω	(2) λ	(3) $\frac{\lambda}{LWL}$	(4) $\Phi^{+}_{\xi\xi}(\omega)$	(5) $N_R(\omega)$	(6) S.M.†	(7) *Product* (4)(5)(6)
1.2	140	4.35	1.39	0.0	1	0.0
1.3	119	3.7	1.18	0.0155	4	0.0731
1.4	103	3.2	0.97	0.0215	2	0.0513
1.5	90	2.8	0.77	0.0310	4	0.0955
1.6	79	2.45	0.61	0.0337	2	0.0410
1.7	70	2.18	0.47	0.0350	4	0.0658
1.8	62.4	1.94	0.38	0.0345	2	0.0262
1.9	56	1.74	0.24	0.0300	4	0.0348
2.0	50.5	1.56	0.23	0.0220	2	0.0101
2.1	45.8	1.42	0.18	0.0155	4	0.0113
2.2	41.7	1.24	0.15	0.0100	2	0.0030
2.3	38.0	1.18	0.12	0.0050	4	0.0024
2.4	35.0	1.04	0.10	0.0	1	0.0

† Simpson multiplier

Σ (7) = 0.4145

$$R^*_{AW} = 2\Delta\left[\frac{\Sigma(7)d\omega}{3}\right]$$

$$R^*_{AW} = 2(27500)\left(\frac{0.4145}{3}\right)(0.1) = 760 \text{ lb}$$

other specifications. In the event that such a model test is not possible, the data and procedures presented in pars. 7-4.1.2 and 7-7 should be used. Considering the scatter in these data, the resulting estimate of power required should be within ± 10 percent of that determined from a model test. In any event, the waterborne power requirements of a new design should be compared to those of similar past designs. This will reveal any unexpected trends and the need for a design review.

7-4.2 LAND OPERATION POWER REQUIREMENTS

The required performance of an amphibian on land is given in the QMR or other vehicle specifications. In the past, the requirements that affect the power required have been given in terms of a required top speed, required grade climbing ability, and possibly, the requirement to tow another vehicle.

As indicated in par. 7-4.1, for almost all amphibians the waterborne performance will require more power than the land performance. The usual design procedure is to choose an engine based on the waterborne power requirements and then determine that the land performance will be satisfactory. The factors involved in the choice of the engine are covered in par. 7-5. Preliminary to the final check of the land performance, information must be available on the tire selected and the arrangement of the

drive train. The selection of tire size will depend on mobility requirements and is covered in par. 7-8. The arrangement of the drive train is influenced by the engine chosen as well as the required tractive effort. The design of the drive train is covered in par. 7-6. The check on land performance is made by determining the required tractive effort as a function of speed and comparing it with the available tractive effort.

7-4.2.1 Required Tractive Effort

The required tractive effort is simply the thrust that must be developed by the tires in order to meet the land performance requirements. This thrust must overcome the resistance of the amphibian to forward motion and provide the force needed to overcome obstacles and grades. The resistance to forward motion is due to the rolling resistance of the tires and the aerodynamic resistance of the amphibian. The rolling resistance of the tire is the work done in compacting the underlying soil, deflecting the tire carcass, and bulldozing soil in front of the tire. Thus, the rolling resistance must be a function of the tire selected and the soild characteristics. Refs. 26 and 63, which are readily available to an amphibian designer, contain a detailed discussion of the rolling resistance and equations for its calculations. The rolling resistance used in the design of the LARC V and LARC XV are given in Table 7-32.

During the engineering tests of the prototype LARC V, tests were conducted to determine the actual rolling resistance and the results of the tests are presented in Figs. 7-58 and 7-59. These data indicate that the rolling resistance varies with soil conditions, tire pressure, and speeds.

In general, the land speed of an amphibian will be low enough that the aerodynamic drag will be small. However, if an estimate of aerodynamic drag is required, it can be obtained from

$$R_{aero} = C_{D_A} \tfrac{1}{2}\rho V^2 A \qquad (7\text{-}68)$$

where

R_{aero} = aerodynamic drag, lb
ρ = mass density of air ≈ 0.00238, slug/ft^3
V = relative speed, ft/sec
A = projected frontal area, ft^2
C_{D_A} = aerodynamic drag coefficient, nd

For typical trucks the value of C_{D_A} is between 0.8 and 2.0. No wind tunnel tests have been conducted to determine the aerodynamic drag coefficient of typical amphibian forms.

The resistance to climbing a grade is given by

$$R_y = W_T \sin \varphi \qquad (7\text{-}69)$$

where

R_y = grade resistance, lb
W_T = vehicle weight, lb
φ = angle of grade with the horizontal, deg

Grades are often given in terms of a percentage, i.e., a 60 percent grade. This notation represents the ratio of vertical distance to horizontal distance, expressed as a percentage.

In the event the amphibian is required to tow equipment, the required tractive effort can be determined from the rolling resistance and weight of the particular item.

7-4.2.2 Available Tractive Effort

The available tractive effort is the maximum thrust that the wheels could develop at a given

TABLE 7-32 DESIGN ROLLING RESISTANCE OF LARC V AND LARC XV

LARC V 4 18.00 × 25 in. tires 12-ply rating
LARC XV 4 24.00 × 29 in. tires 16-ply rating

Conditions	*LARC V*	*LARC XV*
Hard Surface	70-80 lb/ton	70 lb/ton
Cross Country (average)	130 lb/ton	130 lb/ton
Dry Sand at 10 mph	180 lb/ton	
Dry Sand at 25 mph	200 lb/ton	

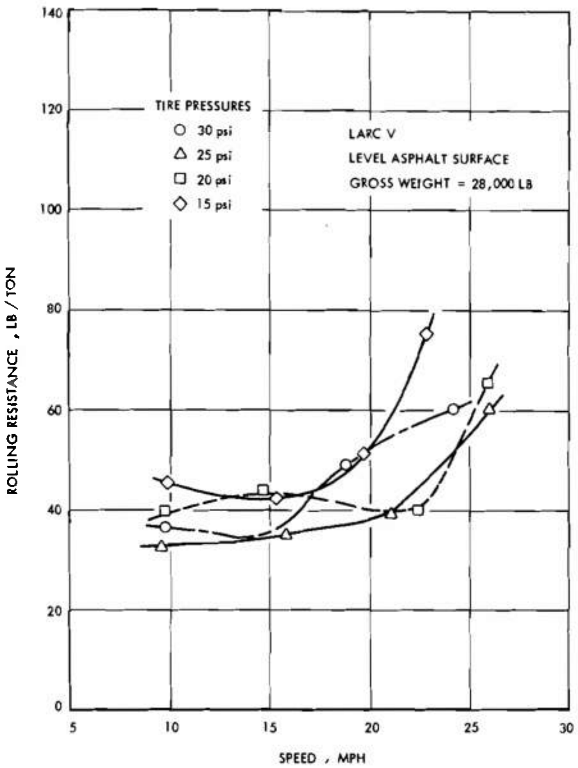


Figure 7-58. Rolling Resistance vs Speed on Level Asphalt Surface

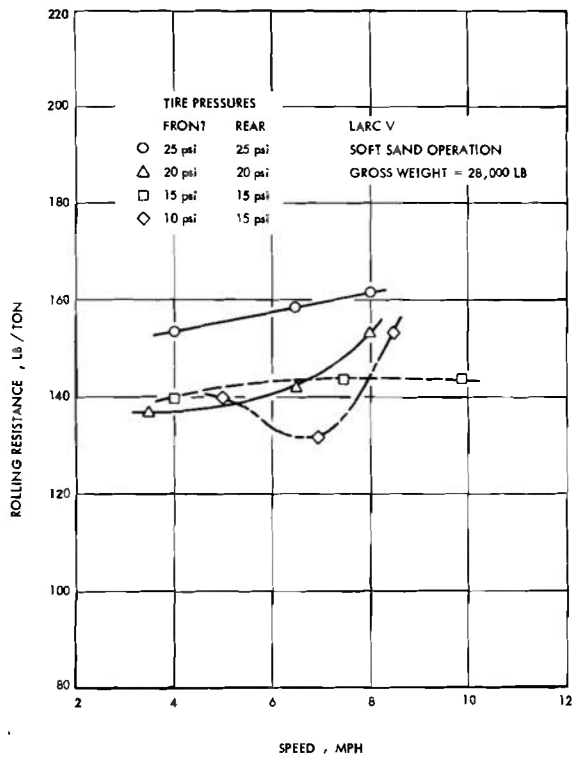


Figure 7-59. Rolling Resistance vs Speed on Soft Sand Surface

speed if the maximum available torque could be absorbed. This is an ideal case since the soil conditions may not allow the maximum available torque to be used.

The calculations of available tractive effort are illustrated in Table 7-33 in a sample calculation for the LARC XV. The engine converter performance curves on which this calculation is based are presented in Fig. 7-60. The resulting tractive effort curves are presented in Fig. 7-61. The required tractive efforts for level ground and for grades are also presented in Fig. 7-61. These curves are based on a rolling resistance of 130 lb/ton which is the assumed cross country average. Under these idealized conditions the LARC XV would have a top speed of 30 mph and a speed of 1.5 mph on a 60 percent grade.

7-4.3 SPECIAL PERFORMANCE REQUIREMENTS, WATER-LAND INTERFACE

In order to serve its function as an amphibious vehicle, the wheeled amphibian must be able to operate in the water-land interface. This ability was shown to be of great importance in World War II and since then in Southeast Asia. The QMR or other specifications for a wheeled amphibian may establish performance requirements on two distinct water-land interfaces, which include surf and river banks. The requirements for each are different and are considered here separately.

7-4.3.1 Surf Operations

The ability of an amphibian to operate through surf is determined by a number of factors in addition to the installed power. These include stability, water-tight integrity, arrangement, and maneuverability and are covered in pars. 7-1, 7-2, and 7-4.

In an operation over a beach the amphibian must pass through the surf coming into and exiting from the beach. In exiting from a beach the amphibian will be headed into the surf and take breaking waves over the bow. The forces developed by a large breaking wave are sufficient

to momentarily stop an amphibian, no matter how much power is installed. This is not important since forward progress will resume as soon as the breaker passes, allowing the amphibian to eventually clear the breaker line.

In crossing surf to reach the beach, there is the danger that the amphibian will be overtaken by a wave as the wave breaks. If the surf is plunging this will result in large quantities of water on the decks as well as the danger of broaching and capsizing. In a plunging surf the crest of the wave breaks in such a way that it falls on the face of the wave in front of the breaking point. In the event that the amphibian does not have a self-bailing cargo deck, the incoming water could sink it. This happened to the SUPERDUCK prototype during surfing trials. Amphibians of the LARC family have been tested in and survived the breaking of 10 to 12 ft high plunging surf. In actual operations it may not be practical to operate amphibians in as high a surf as they can safely survive because of probable damage to the cargo. An amphibian may be designed to have sufficient speed to outrun the breakers, thus avoiding the possibility of being caught in breaking seas. The speed required to do this is, to a first approximation, a function of the breaker height. It is known that a wave slows down as it enters shallow water. The relationship between wave speed, wave length, wave period, and water depth is shown in Fig. 7-62. It is also known

TABLE 7-33 CALCULATION OF AVAILABLE TRACTIVE EFFORT FOR LARC XV

$$\text{Wheel rpm} = 303\,\frac{\text{rev}}{\text{mi}} \times \frac{1}{60}\,\frac{\text{hr}}{\text{min}} \times \text{mph} = 5.05 \times \text{mph}$$

Converter output rpm = Wheel rpm × overall gear ratio
Converter output torque from engine converter match curves
Wheel torque = Converter output torque × overall ratio × efficiency*
Tractive effort = Wheel torque/tire rolling radius

Converter Unlocked					
mph	*Wheel rpm*	*Conv. rpm out*	*Conv. Torque-out, lb-ft*	*Wheel Torque, lb-ft*	*Tractive Effort, lb*
Hi Gear Ratio= 20.302:1					
0	0.0	0	3376	59,000	21,450
2	10.1	205	3000	53,000	19,100
4	20.2	410	2620	46,250	16,700
6	30.3	615	2260	39,900	14,250
8	40.4	820	1960	34,600	12,500
10	50.5	1025	1700	30,000	10,800
12	60.6	1230	1510	26,650	9,620
14	70.7	1434	1350	23,800	8,580
16	80.8	1640	1220	21,550	7,770
18	90.9	1840	1110	19,600	7,070
20	101.0	2050	1000	17,650	6,370
22	111.0	2255	930	16,420	5,920
24	121.0	2460	900	15,900	5,730
26.35**	133.0	2700	840	14,840	5,250

* Power train efficiency from converter to wheels is estimated to be 87 percent.
** Calculated

(cont'd)

TABLE 7-33 (Concluded)

Lo Gear Ratio = 51.105:1					
0	0.0	0	3376	151,700	54,700
2	10.1	515	2450	110,000	36,450
4	20.2	1030	1680	75,400	27,200
6	30.3	1547	1280	57,500	20,650
8	40.4	2065	1010	45,300	16,400
10.46	52.8	2700	840	37,800	13,630
Converter Locked-Up					
mph	*Wheel rpm*	*Conv. * rpm Out*	*Conv.* Torque Out, lb-ft*	*Wheel Torque, lb-ft*	*Tractive Effort, lb*
Hi Gear Ratio = 20.302:1					
17.55	88.6	1800	969.0	17,100	6,170
19.50	98.5	2000	986.0	17,400	6,275
21.50	108.4	2200	980.9	17,300	6,240
23.40	118.2	2400	969.0	17,100	6,170
25.40	128.0	2600	948.6	16,740	6,030
27.30	138.0	2800	923.1	16,300	5,880
29.30	147.8	3000	892.5	15,740	5,680
Lo Gear Ratio = 51.105:1					
6.97	35.2	1800	969.0	43,100	15,520
7.75	39.2	2000	986.0	43,800	15,800
8.52	43.1	2200	980.9	43,600	15,700
9.30	47.0	2400	969.0	43,100	15,520
10.07	50.9	2600	948.6	42,200	15,200
10.85	54.8	2800	923.1	41,100	14,800
11.62	58.7	3000	892.5	39,600	14,300

* Same as engine speed and torque.

that there is a relationship between surf height at the point of breaking and the water depth; this relationship is shown in Fig. 7-63. The data for Figs. 7-62 and 7-63 were obtained from Ref. 64. If a depth-to-height ratio of 0.85 is considered typical, the water depth and, thus speed, can be approximated for a breaker of a given height. The resulting speed is indicated by the solid line on the extreme right of Fig. 7-64. An amphibian with a speed greater than that indicated could outrun the breaker thus avoiding the danger. This, however, is not the only possibility. Since the breaking zone is fixed by water depth, the amphibian may be able to pass through it in the time between successive waves. The success of such a maneuver will depend on the skill and judgment of the operator, and the speed of the amphibian. There is still some minimum speed below which the amphibian could not cross the breaker zone without being caught by the next wave.

It is estimated that the breaker zone that would be troublesome to an amphibian would extend for a distance of 15 breaker heights. Based on this assumption it is possible to establish the minimum speed for an amphibian

that will not be overtaken in the breaker zone, assuming perfect timing on the part of the operator. These minimum speeds for several wave periods are also indicated in Fig. 7-64. Between the needed speed to outrun the breaker and minimum speed needed with perfect timing, is a marginal speed range in which the skill of the operator becomes increasingly important.

The actual operational surf height and speed of the LARC series of amphibians are included in Fig. 7-64, to indicate present practice. The point for the LARC LX indicates the importance of factors in addition to speed in determining surf ability. In this case the large size of the LARC LX relative to the other LARC's increases its surfing ability, despite (or regardless of) its lower speed.

Thus, Fig. 7-64 may be used for guidance in estimating the speed required of a wheeled amphibian in surf operations.

7-4.3.2 Exiting from Rivers

As a result of recent operations in Southeast Asia, it has been realized that the ability to rapidly exit from rivers is of great importance. The transition from water to land as an amphibian climbs a river bank involves a number of complex phenomena both in "terramechanics" and hydrodynamics. Because of the complexity of the problem, at the time of this writing there are no procedures available to the designer to relate river exiting requirements to required power or other vehicle characteristics. A considerable amount of research is being done at this time on the river exiting problem, hence the designer must consult current literature to determine the state-of-the-art. Refs. 65 and 66 contain the results of some of the early research on the problem. These studies provide limited qualitative information for the designer. The most important points seem to be that efficient exiting aids are required (e.g., winches, anchors) and that approaching the bank at as high a speed as possible is very helpful.

7-5 FINAL ENGINE SELECTION

As indicated in par. 6-3.1, it is very unlikely that an engine will be specifically developed for a wheeled amphibian design. As a result, it will be necessary to select an engine from the military or modified commercial engines that are available. In the course of the preliminary and final design, the choice of possible engines must be narrowed down to the one that will result in the minimum cost wheeled amphibian that will meet the QMR or other requirements. A number of criteria that can be applied during the preliminary and final design phases to select the most suitable engine are presented in pars. 7-5.1 through 7-5.2.

7-5.1 GENERAL DISCUSSION

The probable requirements of a wheeled amphibian are such that it should be equipped with an engine which can meet the operational requirement of a tactical or Type I engine, as described in par. 7-14.1 of Ref. 63. These requirements deal with the atmospheric environment in terms of temperature, altitude or barometric pressure, and humidity; and establish longitudinal and lateral slope and waterproofing requirements.

To make a rational engine selection, it is necessary to consider the engine and the drive train as a unit. Only in this way can the combination that will meet the end product requirements at the minimum cost be selected. It would not be impossible to find two engines with similar performance capabilities but different output rpm and torque characteristics. In this case one of the engines is likely to be more compatible with a drive train using standard or available components and, accordingly, would be the better choice. The considerations and criteria that apply to the design of the drive train are given in pars. 7-6 through 7-6.5.

7-5.2 CRITERIA FOR WHEELED AMPHIBIAN ENGINES

A number of general criteria which apply to engines of all military vehicles, including wheeled amphibians, include reliability, durability, maintainability, availability, efficiency, weight, and economy. Since these criteria are common to the engine selection for all Army vehicles, they are covered in detail in Ref. 63 and need not be discussed further here. There are, however, several criteria that are unique to wheeled amphibians and the circumstances in which the application to a

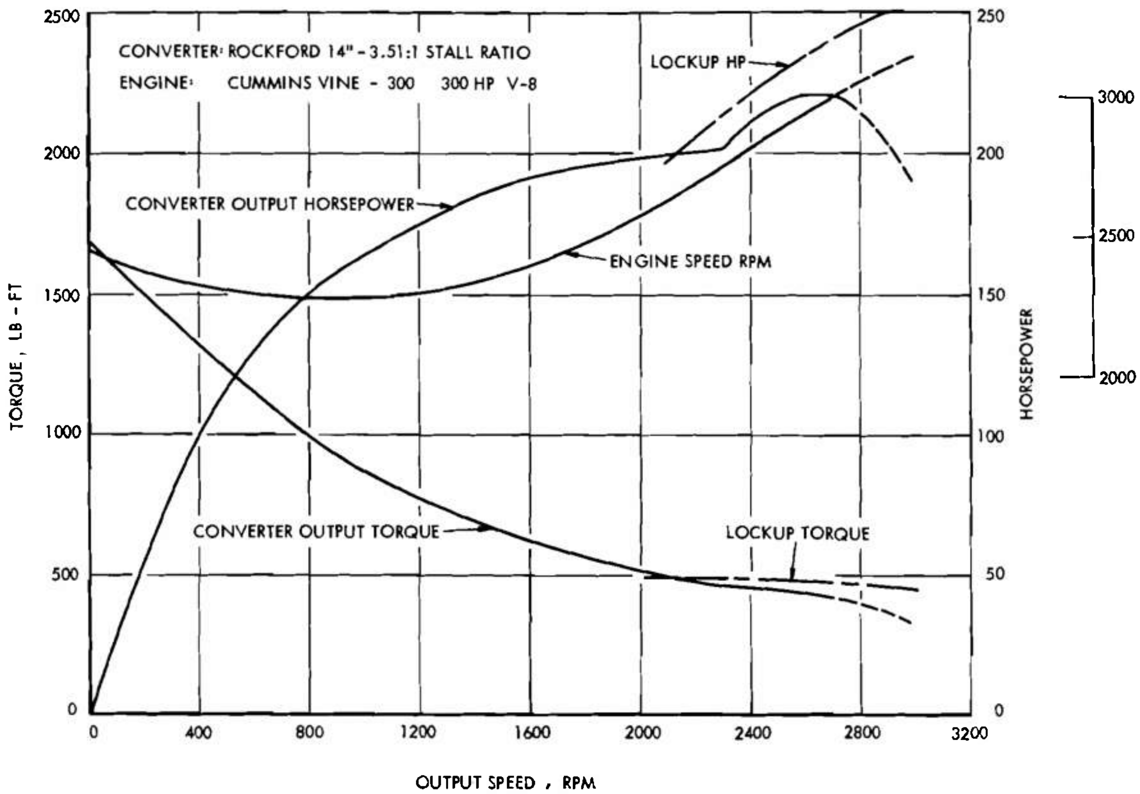


Figure 7-60. Engine-converter Performance

wheeled amphibian is unusual. These cases are presented in pars. 7-5.2.1 through 7-5.2.3.

7-5.2.1 Engine Rating

During waterborne operation, the engine of a wheeled amphibian will be required to deliver maximum power continuously. This condition is more demanding of an engine than any automotive application, where the engine need not deliver maximum power on level ground and down hills. As a result an engine selected for a wheeled amphibian must be capable of delivering the required power for a high percentage of its life, as discussed in pars. 7-4 through 7-4.3.

In establishing the required power the designer must remember to include the load for the auxiliary systems as well as the main propulsion system. This includes the power for the electrical, hydraulic, and cooling systems which in some cases may be as high as 10 percent of the total.

In comparing the required power with the rated power of the engine, the designer must determine the corrections to the rated power for the environmental condition expected. This is most important in the case of the temperature of the inlet air. If the engine takes air from the engine compartment, the temperature may be considerably above that outside the vehicle.

In general, the power rating of an engine and its reliability, life, and maintenance requirements when run at that rating are determined by test. During such tests the engine is run through a duty cycle which includes a certain percentage of the cycle at rated power and the remainder of the cycle at different fractions of the rated power. This test cycle for U. S. Army qualified Diesel engines is given in MIL-STD-1400, and engines that have passed this test are listed in Qualified Products List (QPL) 11276. The U. S. Navy also performs an endurance test to establish the ratings of Diesel engines as given in MIL-E-2357A (Ships). The Navy test reflects the service of an engine in a Navy boat and may be more applicable to wheeled amphibians than the Army test. In the

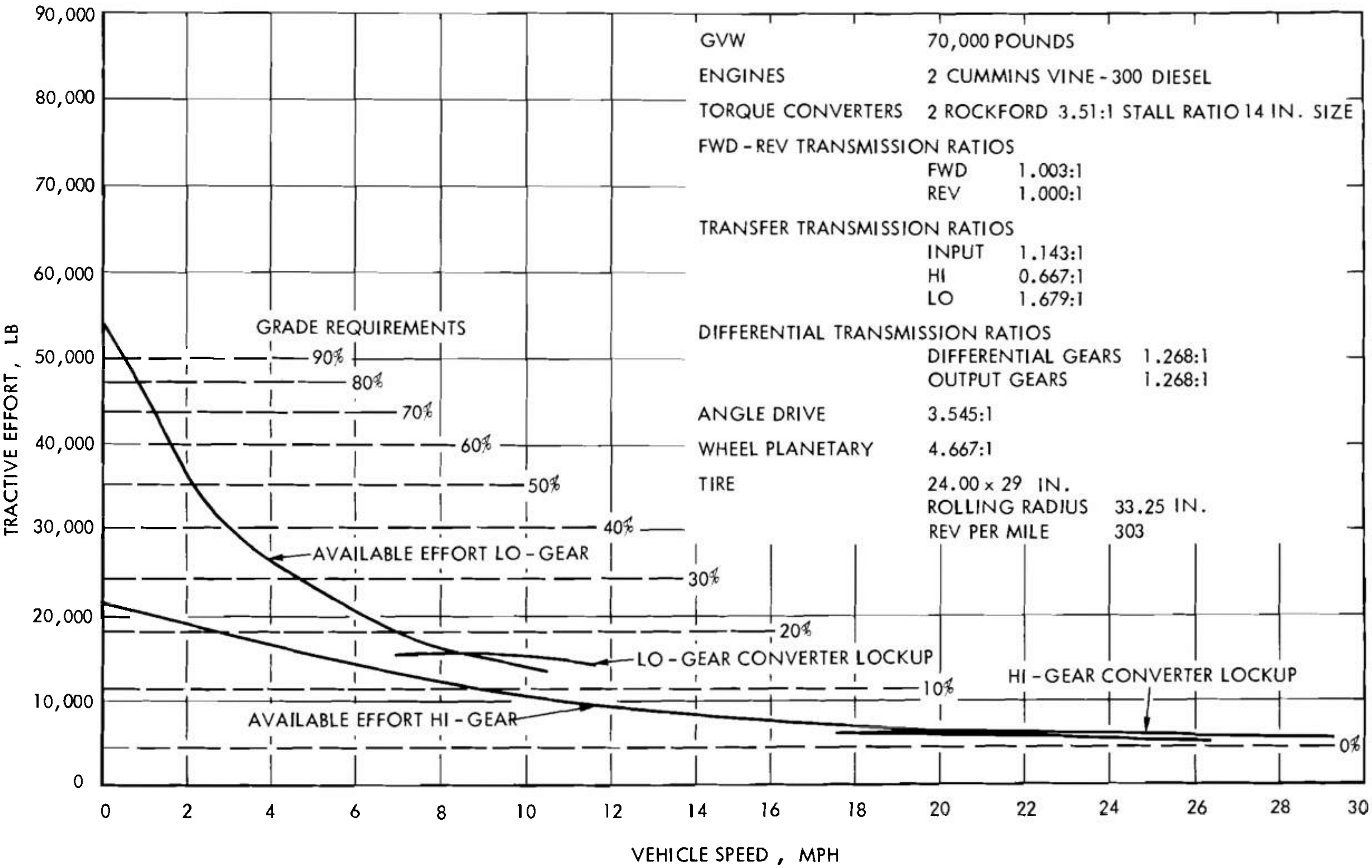


Figure 7-61. Predicted Land Performance of LARC XV

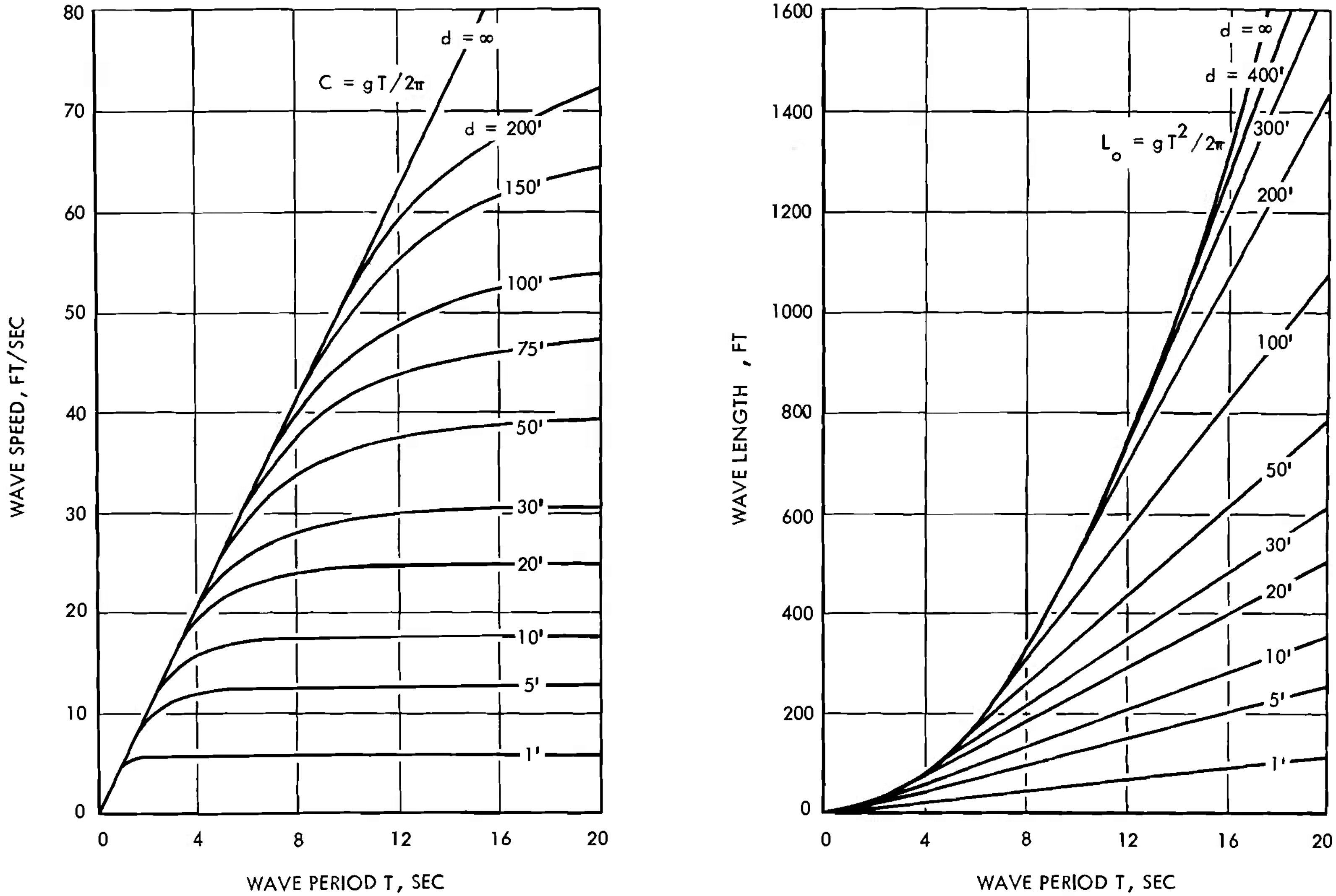


Figure 7-62. Relationships Between Wave Period and Wave Speed and Between Wave Length and Water Depth

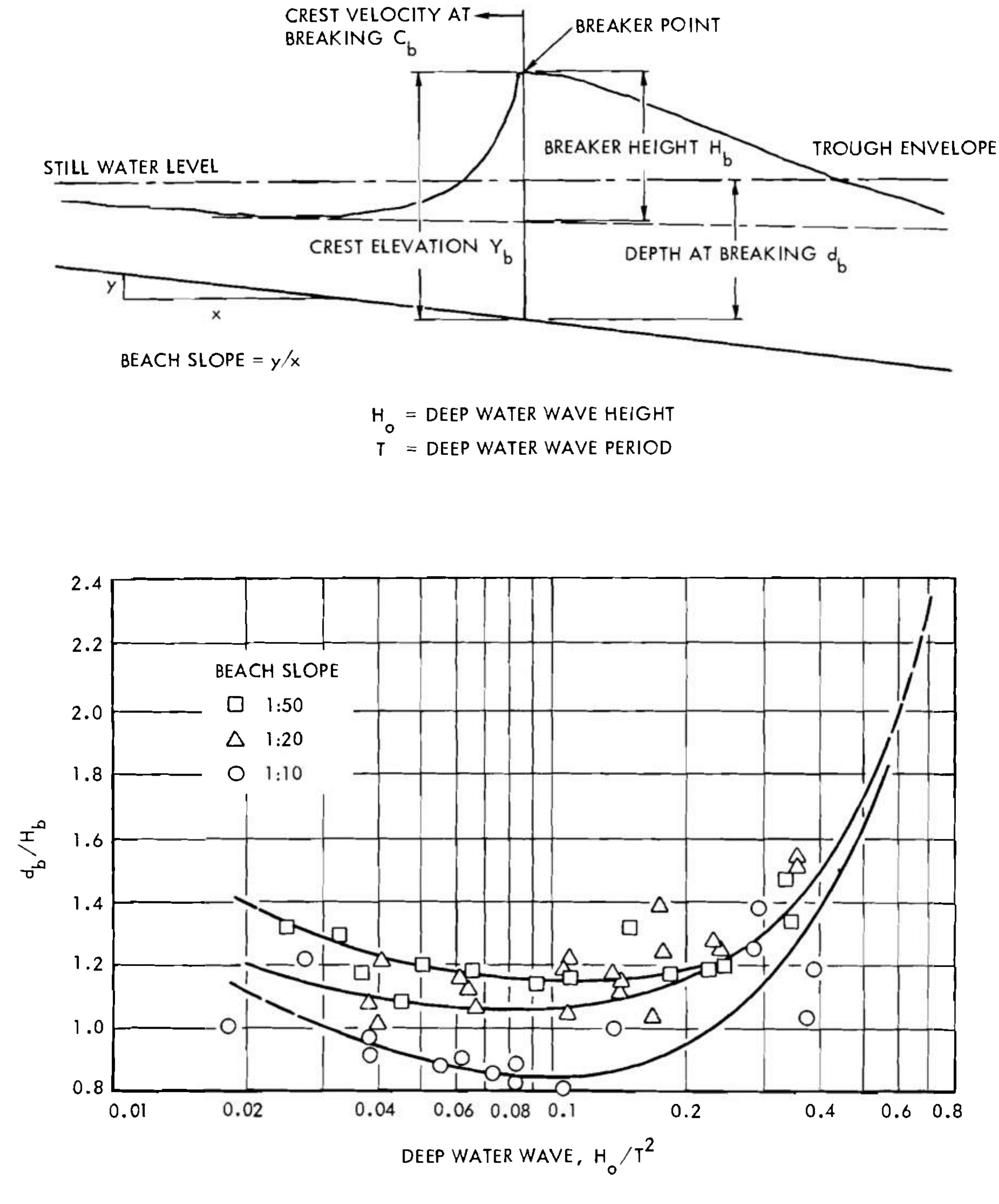


Figure 7-63. Breaker Height vs Water Depth

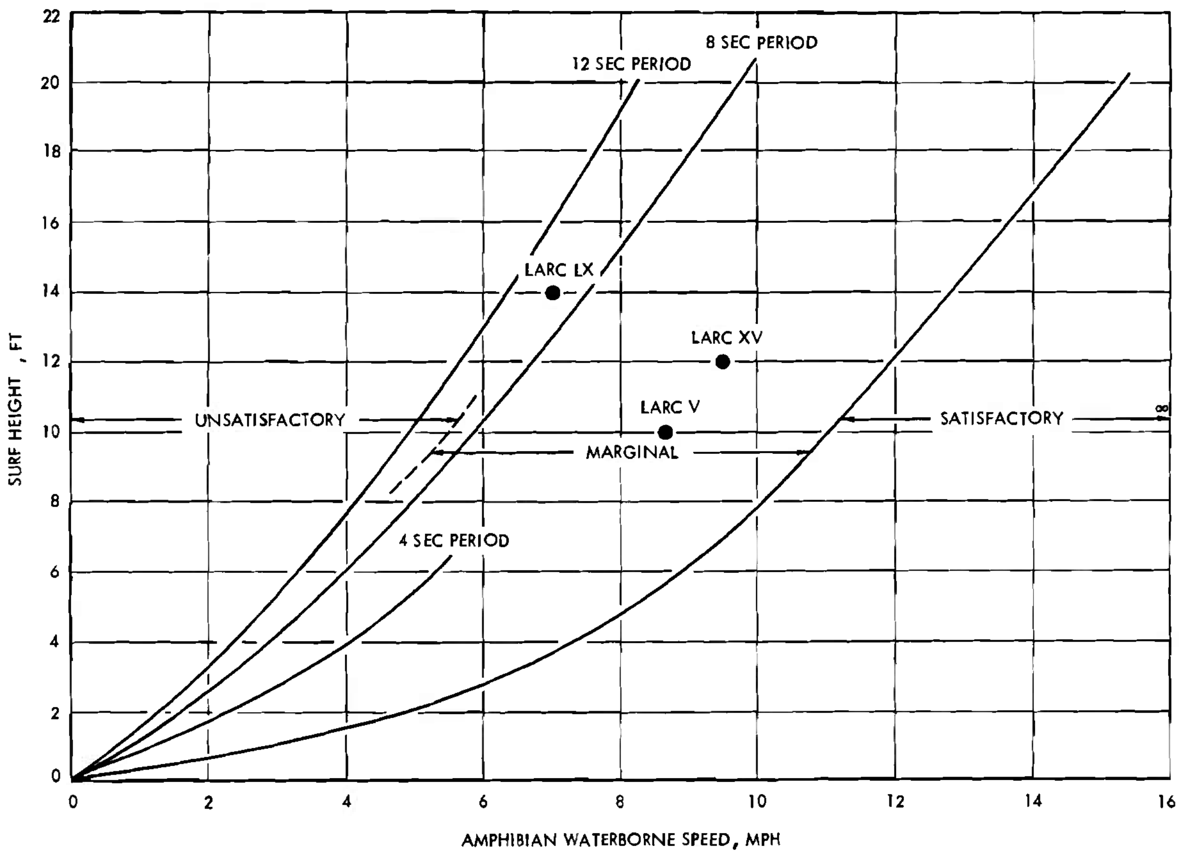


Figure 7-64. Surf Operation Diagram for Wheeled Amphibian

event that these Army and Navy tests do not provide enough information to the wheeled amphibian designer, a candidate engine can be tested with any desired duty cycle. The U. S. Army Tank-Automotive Command Propulsion Systems Laboratory at Warren, Michigan, has the facilities for such a test.

7-5.2.2 The Amphibian Environment

The environment in which the wheeled amphibian engine must operate imposes certain requirements on the selection of the engine. The first such requirement is the tolerance of the engine to salt water. An engine in an amphibian must be capable of withstanding the effects of being wet with salt water and of ingesting salt water or salt particles in the air intake. After an amphibian has been in service for some time, it would not be unusual for the hatches over the engines to leak in rough water and surf. This requires that the engine be waterproofed in accordance with the requirements referenced in par. 7-5.2. It should be noted with respect to the engine's electrical system that salt water is a much better conductor than fresh water.

The problem of salt water or salt particles in the intake air is much more serious than the problem of direct wetting if the engine is properly waterproofed. This is particularly important for gas turbine engines where compressor fouling and sulfidation are involved. Compressor fouling occurs when salt water droplets in the intake air impinge at high velocities on compressor blades and stators. The water scatters and is evaporated. The salt crystals adhere to the blade surfaces and to each other, thus distorting the airfoils and lowering component efficiency. Additional fuel must be provided to restore power, causing the engine to run hotter and closer to surge condition. The engine power may be restored simply by washing the compressor with fresh water and, accordingly, a gas turbine in a wheeled

amphibian must be fitted with such a fresh water wash system (see par. 7-10.10).

In order to reduce the number of fresh water washings required and increase the life of the engine, the amount of salt water and salt particles ingested in the intake air must be minimized. Several types of separators have been developed, including mesh, inertial, and electrostatic types. These can be designed to remove particles of 5-micron diameter and larger with flow velocities between 10 and 75 ft/sec. The electrostatic type can be overloaded if the amount of salt in the air exceeds 10 parts per million by weight. The U. S. Naval Boiler and Turbine Laboratory has been developing separators for Naval vessels and should be contacted by the amphibian designer for the latest developments.

Sulfidation is a much more serious problem with respect to gas turbine engine life. At temperatures above 1,550°F the sulfur in the fuel combines with salt in the intake air to form a slag which is very corrosive to nozzle guide vanes and turbine blades fabricated from nickel base and cobalt base superalloys. Unprotected nickel base alloy blades that have been run in tests in a marine environment at temperatures above 1,550°F have failed in less than 200 hr. Metallic coatings may be used to overcome this corrosion problem. However, to date coatings have not been developed that will allow a time between overhaul equal to that of the same engine in a nonmarine environment.

The amphibian designer should be cautious in the choice of an automotive or aircraft gas turbine in an amphibian application unless the engine has been modified and tested for use in a marine environment. In no case should such an engine be the final selection until such modifications have been made and the engine subsequently tested to the satisfaction of all criteria.

Safety with respect to fire and explosion hazards is a major engine selection criterion. The hull of an amphibian is closed so that any fuel leakage from the engine or fuel system will collect in the bilge. If the flash point of the fuel is sufficiently low, the hull will fill with vapor in sufficient quantity to form an explosive mixture. Thus a gasoline engine should not be selected for a wheeled amphibian unless no suitable engine that burns Diesel oil is available. The design characteristics of a fuel system to avoid fire and explosion hazards are covered in par. 7-10.3.

7-5.2.3 Weight, Volume, and Cost Effectiveness

In the course of the preliminary and final design phases, the number of possible engines for a given design can be narrowed by use of the criteria given in pars. 7-5.2, 7-5.2.1, and 7-5.2.2. In many cases this will result in only one engine with suitable characteristics. This was the case with the selection of the Diesel engines for the production LARC V and LARC XV. The designer should note that multiple engine installations are more practical for a wheeled amphibian than for a conventional vehicle. This fact affects the number of engines that must be considered in designing an amphibian. In the event that two or more engines are available with suitable performance characteristics, the final selection should be made on a cost-effectiveness basis. This is not to imply that the lowest cost engine should be selected. Rather the engine which results in the lowest life cycle cost for the amphibian should be chosen. This requires the consideration of the engine as well as the drive train, fuel capacity, auxiliary systems, and interactions with the hull. In this way, it is possible to make a rational choice between engines with differing initial costs, weights, volume, and fuel consumption. This procedure may result in an increase in the design costs, especially if the engines have output characteristics different enough to require the design of different drive trains. However, the design costs of a wheeled amphibian are a very small percentage of the total life cycle cost. Thus a large increase in the design costs are justified, even if they result in a small percentage reduction in procurement or operational costs.

7-6 POWER TRAIN DESIGN

An extensive introduction to the subject of Army automotive power trains is presented in Ref. 63. The limited space available in this handbook makes it possible to present information only on the differences between wheeled amphibian power trains and those used on conventional Army wheeled vehicles.

7-6.1 GENERAL DISCUSSION

The function of the power train on a wheeled

amphibian is the transmission of power from the engine to the wheels and water propulsor. The engine torque and rotational characteristics must also be modified by the power train to suit the wheels and water propulsor requirements.

A major design consideration in both the preliminary and final design involves the choice of standard military or commercial components, or the development of custom components. In the preliminary design phase this decision involves major components such as transmissions and transfer cases and, in the final design phase, minor components such as propeller shafts. In both the preliminary and final design phase the selection depends on considerations of performance, arrangement, weight, reliability, maintainability, and cost. The advantages of standard components with respect to reliability, maintainability, and cost are discussed in Chapter 2. However, requirements for performance, arrangements, or weight may prevent the use of standard components. Pars. 7-6.2 through 7-6.5 present the characteristics, criteria, and functions that are unique to wheeled amphibian power trains. This information may be used for the selection of the correct standard components or in the design of custom components.

7-6.2 CHARACTERISTICS AND CRITERIA

In order for a wheeled amphibian to operate successfully with maximum effectiveness, the power train must have certain characteristics, as discussed in the paragraphs which follow.

7-6.2.1 Necessary Power Train Characteristics

a. Flexibility in Power Distribution. A wheeled amphibian must be able to apply power to either the marine propulsor or wheels separately, or to both simultaneously. The ability to selectively power the marine propulsor or wheels is needed for efficiency when operating on land or water. This is most important in water operations since rotating the wheels wastes a great deal of power. The marine propulsor and the wheels must both be powered when the amphibian enters or exits from the water to maintain the necessary speed in order to reduce the danger of broaching in the surf or bogging down on a river bank.

b. Durability. In addition to the usual loads and shocks that the power train of a conventional wheeled military vehicle must withstand (see Ref. 63), the power train of a wheeled amphibian must withstand loads and shocks peculiar to its environment. In particular, in crossing the surf zone or approaching a river bank the wheels must be engaged to provide quick acceleration onto the beach or bank. If we assume a mechanical drive, the wheels will turn at some ratio of the propulsor speed. Contact with the beach or bank will result in high shaft and transmission torque due to the momentum change and the transfer of power to the more resistive medium. This condition is even more serious if a free turbine type gas turbine engine is used. In this case the energy change due to the sudden deceleration of the power turbine must be dissipated.

c. Controllability. When a wheeled amphibian approaches a beach through surf, the driver will be under considerable stress. As a result, the control of the power train must be as simple and foolproof as possible. In particular, there must be no chance that the wheel drive can be put in reverse. The resulting stop when the wheels make contact could cause the vehicle to broach or bog down.

7-6.2.2 Desirable Power Train Characteristics

A number of desirable characteristics are useful in a wheeled amphibian power train to enhance its effectiveness. The extent to which these desirable characteristics are included in a given power train design should be determined by a trade-off study which considers the entire vehicle. Increases in effectiveness must be traded off against any increases in the life cycle cost of the vehicle. Desirable characteristics for a wheeled amphibian power train include the following:

a. High Transmission Efficiency for Waterborne Propulsor. The marine propulsor will probably have a much larger power demand than the land propulsor. As a result, considerable savings in engine power and fuel consumption may be possible if the marine part of the power train has a high efficiency. In particular, the efficiency of a drive train with a torque converter can be increased from about 85 percent to above 95 percent by including a mechanical "lock-up" with the converter. The control of the mechanical lock-up can be either

manual or automatic.

b. Independent Marine Propulsor and Wheel Speeds. When approaching a beach or river bank it is useful to be able to adjust the rotational speed of the marine propulsor and the wheels independently. This will reduce the landing shock in the drive train and reduce the chance of bogging down during the water-land transition. This has been accomplished in the LARC V and LARC XV by arranging the transmission so that the land drive can be placed in the high or low range without affecting the speed of the marine propulsor.

c. Simplicity in Automotive Transmission. Because of the high power installed in wheeled amphibians, the automotive transmission requires fewer gear ratios than comparable equipment in trucks. Accordingly, a less complex automotive transmission can be provided for an amphibian making it simpler to drive on land. For example, the LARC V and XV have only 2 forward speeds whereas a military truck may have 5 or more. However, if a suitable standard transmission is available it may be cost effective to accept its complexity instead of developing a simpler new unit.

d. Independent Control of Multiple Marine Propulsors. Large wheeled amphibians may be equipped with twin marine propulsors. In this case a significant gain in waterborne maneuverability can be obtained if the marine propulsors can produce thrust in opposite directions. This feature is most useful for maneuvering alongside a cargo ship and may reduce loading time.

7-6.2.3 Power Train Criteria

In general, the same criteria that apply to the selection or design of power trains for conventional wheeled military vehicles apply to wheeled amphibians. These criteria are covered in Ref. 63. There are, however, several criteria that are of particular importance to a wheeled amphibian and are similar to those that apply to the engine selection, as presented in pars. 7-5.2 through 7-5.2.3. These criteria, as they apply to wheeled amphibian power trains, are presented in pars. 7-6.2.3.1 through 7-6.2.3.3.

7-6.2.3.1 Power Ratings

The life of the power train components must be compatible with the requirements given in the QMR. Those components that supply power to the marine propulsor must be able to transmit the high power required for a high percentage of their life. This could be a serious problem if an automotive transmission were used to provide power to both the marine propulsor and the wheels. The low gear ratios in such transmissions may not have sufficient life for the marine propulsor requirements.

7-6.2.3.2 The Amphibian Environment

It is probable that the power train components will be located low in the amphibian hull and, as a result, will be periodically wetted with bilge water. Accordingly, the components must be waterproofed and made of corrosion resisting materials. As a minimum, the components should be protected with proper paint or coatings.

7-6.2.3.3. Weight, Volume, and Cost Effectiveness

The final selection or design of the power train must depend on obtaining the most cost-effective vehicle to meet the requirements. This involves the same considerations as govern the engine selection and are covered in par. 7-5.2.3.

7-6.3 THE MARINE POWER TRAIN

The function of the marine part of the power train is the transmission of power at a suitable rotational speed to the marine propulsor. The typical elements of the marine power train, with their functions or characteristics, as applied to wheeled amphibians, are considered in the paragraphs which follow.

7-6.3.1 Transfer Case

In a mechanical drive train, the transfer case receives power from the drive train and delivers it to the marine propulsor shaft. The transfer case may provide any needed change in rotative speed and displacement of the output shaft needed to suit arrangements. The transfer case may include a clutch to disengage the propeller for land operation. The transfer case must transmit the full power for long periods and must be designed for adequate life. It must also be designed to withstand the maximum torque

that will result if the marine propulsor is suddenly stopped by floating debris.

7-6.3.2 Propulsion Motor

In a hydrostatic or electric drive train, a propulsion motor will supply power directly to the marine propulsor shaft. This motor serves the same purpose as the transfer case in the mechanical system and must satisfy the same criteria.

7-6.3.3 Propulsor Shaft

The propulsor shaft transmits the output torque from the transfer case or propulsion motor to the marine propulsor. The propulsor shaft may include a slip clutch or shear pin to limit the torque in the drive train if the propulsor is fouled by debris. Considerable savings in maintenance time may be realized if the propulsor shaft is arranged in a manner to permit rapid removal. This will allow the rapid changing of a damaged propeller or other associated components.

7-6.3.4 Bearings and Seals

The bearings and seals support the propulsor shaft and provide a seal where the shaft penetrates the hull. On small boats the bearings outside the hull are water lubricated. This type of bearing is not suitable for a wheeled amphibian since the marine propulsor may be run when out of the water. To save wear on all the shaft bearings, a shaft lock can be provided to keep the propulsor shaft from free-wheeling when the marine drive is declutched. Before doing this, it must be determined that serious wear will not occur on the marine clutch due to drag against the locked shaft.

7-6.3.5 Marine Propulsor

The marine propulsor converts engine power into useful thrust to propel the amphibian in water. Design of the marine propulsor is covered in pars. 7-7.1 through 7-7.6.4.

7-6.4 THE AUTOMOTIVE POWER TRAIN

The function of the automotive part of a wheeled amphibian drive train is the transmission of power from the engine to the wheels and the marine transfer case or drive motor. The automotive drive train modifies the rotational speed and torque as required for the wheels and provides reversing for the wheels and, possibly, for the marine propulsor. The types and typical elements of the automotive power train, with their functions or characteristics that apply especially to wheeled amphibians, are considered in the discussion which follows. Those functions and characteristics that apply to conventional wheeled vehicles as well as amphibians are discussed in Ref. 63.

7-6.4.1 Power Train Types

Three basic types of power trains can be used on a wheeled amphibian—mechanical, hydrostatic, and electric. Each type has certain advantages and disadvantages for use in a wheeled amphibian. These are presented in pars. 7-6.4.1.1 through 7-6.4.1.3. The selection of the power train type should be made on the basis of the characteristics and criteria presented in pars. 7-6.2 through 7-6.2.3.3.

7-6.4.1.1 Mechanical

In a mechanical power train there is a direct mechanical connection through gears and shafts between the engine, and the wheels and marine propulsor. Mechanical power trains have been used on all wheeled amphibians built to date as well as on the vast majority of all wheeled military vehicles. As a result, there exists a wide variety of standard components familiar to maintenance and operating personnel. The advantages that result from the use of standard components are presented in Chapter 2.

Further advantages of a mechanical drive include a high transmission efficiency and light weight relative to the other drive train types. High transmission efficiency is very important for the parts of the drive train that supply the marine propulsor since they must transmit high power for long periods. Improvements in the efficiency of the marine drive will thus appear directly as reductions in engine power and fuel consumption. If the amphibian has multiple engines, the power train should be arranged so that one engine alone can propel the amphibian in emergencies.

The disadvantages of a mechanical drive train include a lack of flexibility in power distribution between the wheels and marine propulsor, and a lack of flexibility in arrangements. This latter factor is of importance in high performance wheeled amphibians which may be fitted with retractable wheels and propulsors.

7-6.4.1.2 Hydrostatic

In a hydrostatic power train, the power is transferred from the engine to the wheels and marine propulsor by the flow of hydraulic fluid under static pressure. The pressure and flow is supplied by a pump driven directly by the engine and converted to useful work by hydraulic motors connected directly to the wheels or marine propulsor. Hydrostatic drive trains have been used on small commercial wheeled vehicles and in experimental military vehicles. Research and development in this area have been carried out by the U. S. Army Tank-Automotive Command Propulsion Systems Laboratory at Warren, Michigan. It is of interest to note that a hydrostatic drive train was fitted as an experiment in one of the prototype LARC XV's in the early 1960's. Unfortunately a fire destroyed the vehicle before the test program was completed. The results at the time of the fire indicated that the system was practical.

The advantages of a hydrostatic drive train in a wheeled amphibian include great flexibility in power distribution and in physical arrangements. In a hydrostatic drive train, power can be transmitted to retractable wheels and marine propulsors without the use of complex shafts, gear trains, and universal joints. A hydrostatic drive train may also allow high engine efficiency to be obtained in land operation. The engine can be automatically controlled to run at its most efficient speed under all circumstances. Vehicle speed control is obtained by regulating the pump output. This may be a particular advantage in the case of gas turbine engines since their efficiency and life are improved if rapid speed changes are avoided.

The disadvantages of a hydrostatic drive train include lower transmission efficiency than a mechanical drive train, probability of higher weight, and fewer available standard components. These disadvantages are likely to be overcome by the rapidly advancing state-of-the-art. As a result, a hydrostatic drive train should be seriously considered for a wheeled amphibian if the number of vehicles to be built justifies the necessary engineering and design work.

7-6.4.1.3 Electric

Electric drive trains have been used in a wide variety of vehicles ranging from electric automobiles to railroad locomotives and ships. These applications have included experimental military vehicles and very large gas turbine powered earth movers. Research and development work has been carried out on electric drive trains by the U. S. Army Tank-Automotive Command Propulsion System Laboratory at Warren, Michigan, and the U. S. Army Mobility Equipment Research and Development Center at Fort Belvoir, Virginia.

The advantages of an electric drive train are similar to those of a hydrostatic drive train and include flexibility in power distribution, arrangements, and control for maximum engine efficiency and life.

The disadvantages of an electric drive train are also similar to those of the hydrostatic drive train and include relatively low efficiency, high weight, and a lack of available standard components. If the number of vehicles under consideration justifies it, the engineering and design time required should be devoted to the serious study of an electric drive train for a wheeled amphibian.

7-6.4.2 Power Train Components

7-6.4.2.1 Torque Converter

A torque converter is required in a mechanical automotive drive train of a wheeled amphibian powered with a reciprocating internal combustion engine or single shaft gas turbine to provide the necessary speed-torque characteristics. If the torque converter also transmits power for the marine propulsor, it should be equipped with a mechanical lock-up feature to improve the marine power train efficiency. The lock-up should be controlled so that it occurs only in the marine mode. In a wheeled amphibian, when power is supplied to the wheels or wheels and marine propulsor, torque converter lock-up is not required or desirable.

7-6.4.2.2 Transmissions and Transfer Cases

In a mechanical automotive drive train, the transmission and transfer case provide the required changes in rotative speed, direction, and displacement of power output shafts. In the LARC V and LARC XV, two separate transmissions are provided—one supplies forward, neutral, and reverse to the wheels and the marine propulsor; and the other provides speed ratios to the wheels only. In this way the required flexibility for beaching is obtained with a minimum number of components. There are a number of transmissions available for conventional wheeled vehicles that may be considered for use in a wheeled amphibian. In general these combine a torque converter, forward, neutral, and reverse, and a number of gear ratios into one unit. If such a transmission supplies power to the wheels only with a separate forward, neutral, and reverse transmission for the marine propulsor; no special problem occurs. This was the arrangement used on the LVH-X2 high-speed wheeled amphibian. However, if such a transmission supplies both the wheels and marine propulsor, it must be designed for life at the high marine power levels and peak torques that may occur from a jammed marine propulsor. If a gas turbine engine is used, the peak torques that can be developed on a cold day must also be considered.

7-6.4.2.3 Differentials, Drive Shafts, and Universal Joints

These components serve the same functions on a wheeled amphibian as on a conventioanl wheeled vehicle. However, if the amphibian has a rigid suspension system, unusual loads are applied to these components. The great torsional stiffness of an amphibian hull combined with a rigid suspension results in the certainty that in rough terrain it will occasionally be suspended on diagonally opposite wheels, as illustrated in Fig. 7-65. The amphibian must be propelled when suspended in this manner. The simplest and most direct way to accomplish this is to drive the wheels on each side with a common shaft, shown in Fig. 7-65. A single differential is provided between the drives on each side of the amphibian. The conventional truck arrangement with limited slip differentials between each pair of wheels will not provide as good mobility since the limited slip differentials can only support about a 3 to 1 torque difference. If the arrangement shown in Fig. 7-65 is used; the differential, shafts, gears, and universal joints must be designed to take the high torque that can occur in the suspended condition. These torques are roughly twice the value that would occur in the drive train of conventional wheeled vehicles of similar weight, power, and running gear arrangement. In some arrangements the drive shafts may become long enough that torsional vibrations are a problem. Stiffer shafts or vibration dampers may then be required.

7-6.4.2.4 Pumps, Motors, and Generators

These components are used in hydrostatic and electric drive trains to take the place of the mechanical components listed in pars. 7-6.4.2.1 through 7-6.4.2.3. The design, selection, and arrangement of these components for a wheeled amphibian drive train are similar to those for a

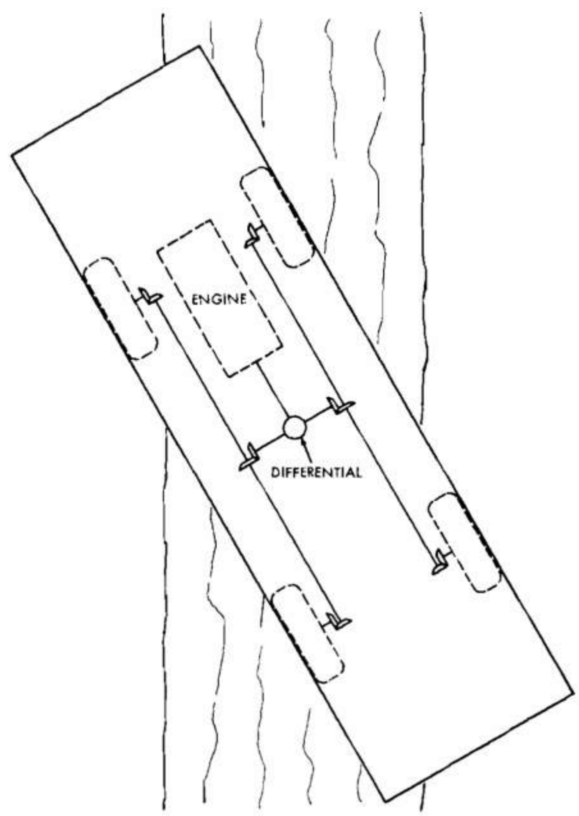


Figure 7-65. LARC-type Land Power Train

conventional wheeled vehicle as covered in Ref. 63. It should be remembered that the hull of a wheeled amphibian is completely enclosed so that heat buildup could be a problem. As a result, a cooling system for the drive train may be necessary.

7-6.5 BRAKES

A discussion of the theory of retardation, types of brakes and brake mechanisms, and systems as applied to conventional wheeled vehicles is presented in Ref. 63. In general, the same brake mechanisms and systems can be applied to wheeled amphibians. There are, however, some special considerations imposed by the amphibian environment.

The brakes used on the WW II DUKW were of the internal-expanding type mounted on the wheel. When the vehicle was driven into the water, the rapid cooling of the brakes that were hot from land operation created a partial vacuum in the brake housing. This combined with the hydrostatic pressure would fill the brake housings with water despite all attempts to seal them. The resulting corrosion and damage caused by sand carried in with the water caused failures and a tremendous amount of maintenance work.

As a result of this experience, all production wheeled amphibians since then have been fitted with brakes on the drive shafts inside the hull. These brakes have been of the disk type. The major problem with locating the brakes in the hull is dissipating the heat generated and maintaining the brake temperature at an acceptable level. This problem can be solved by providing an adequate air flow through the hull during land operations.

As an example of the amounts of energy that must be dissipated, consider the case of stopping an amphibian on a 20 percent grade from a speed of 20 mph with a deceleration of ½ g. The energy dissipated is given by

$$E = \frac{(Mb - F)V_o^2}{2b} \tag{7-70}$$

where

E = total energy, ft-lb
M = mass of vehicle, slugs
b = deceleration, ft/sec^2 (consider positive)
V_o = initial speed, ft/sec
F = external forces on vehicle, lb (rolling resistance—grade effect)

If the gross weight is 30,000 lb and the rolling resistance is 40 lb/ton the total energy dissipated is 715,000 ft-lb or 920 BTU. This energy is dissipated by the brakes in the time it takes to stop the vehicle which is 1.82 sec. As a result, the temperature of the brakes rises which causes a transfer of the energy to the surrounding air. The breaking forces and the rate at which the brakes must dissipate energy must be determined on the basis of the performance requirements given in the QMR.

In addition to brakes, some wheeled vehicles have been fitted with hydraulic retarders to dissipate energy. These retarders dissipate mechanical energy in hydraulic fluid in the form of heat which is, in turn, rejected by the fluid in a cooler. Hydraulic retarders are often combined with the torque converter. A retarder has the advantage of reducing wear on the brakes. A disadvantage is the need to include additional equipment (i.e., the retarder and oil cooler) in the vehicle. This is a serious problem in an amphibian where space and weight are at a premium.

7-7 MARINE PROPULSOR DESIGN

7-7.1 GENERAL DISCUSSION

The resistance of a ship to motion through water is overcome by an equal and opposite propelling force. The energy for propulsion in an amphibian is supplied by an internal combustion engine, either a Diesel or gas turbine. This energy is transmitted to the amphibian as a propulsive force by the marine propulsor. This chapter discusses: (1) the criteria upon which the selection of the type of propulsor is made, (2) the advantages of various types of propulsors, (3) the basic theory of marine propellers, (4) the design of screw propellers, and (5) the governing laws and application of model tests.

7-7.2 CRITERIA FOR PROPULSION UNITS

A large number of factors must be considered in the selection of the type of marine propulsion system for an amphibian. Once the type of unit has been selected, these same factors will enter into the details of the particular design. The

relative importance of each factor will depend on the particular requirements specified in the QMR. Factors to be considered include, but are not limited to: maintenance, simplicity, durability, vulnerability, reliability, availability, efficiency, weight, arrangements, effectiveness, and economy. These factors are not independent. For example, ease-of-maintenance, reliability, and economy are all associated with simplicity. Also, the factors may not always be applied to the propulsion unit alone but must consider the amphibian and its propulsor as a system. These factors are discussed in par. 5-1.3.

7-7.3 PROPELLER TYPES

The selection of the marine propulsor type for an amphibian will be affected by special considerations such as low speed maneuverability, and general criteria such as cost and dependability. No particular type will exhibit all of the desired qualities. The designer must consider the trade-offs that result from conflicting requirements. For example, momentum theory shows that efficiency increases with propeller diameter whereas the shallow draft requirements may favor the selection of a small propeller which can be protected from damage while the amphibian is passing through the surf. Some alternative solution may satisfy both objectives (shallow draft and high efficiency), but with penalties in terms of added complexity or cost. Two small propellers may be just as efficient as one large propeller and can have an added premium in maneuverability. The propellers on the high-speed amphibian LVW are supported by a plate which retracts into the hull so that the propellers are less vulnerable to damage in the landing configuration. Contra-rotating propellers could be used to achieve relatively high efficiency with small diameter. Kort nozzles (see par. 7-7.3.2) have been installed on many existing amphibians because they increase the effective area of a propeller and affort a good deal of protection.

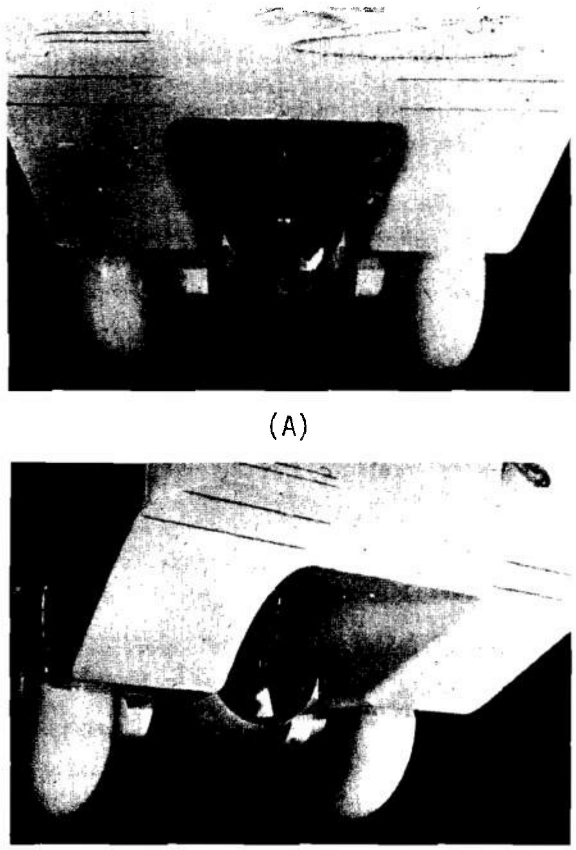
The type, size, location, and special features of the marine propulsor can all affect the basic hull geometry and arrangements of the amphibian and should be selected in the early stages of the design to assure compatibility between the hull and propeller. This section presents various types of propellers which may be suitable for amphibians. The advantages and shortcomings of each type will be discussed.

7-7.3.1 Screw Propeller in Tunnel

The screw propeller is the most common marine propulsor. It is a relatively efficient thrust-producing mechanism when properly designed and installed on a suitable hull. Open water efficiencies on the order of 80 percent of ideal efficiency can be realized and propulsive coefficients as high as 0.70 are common for conventional ships. A great deal of series data has been collected, and design procedures and charts have been developed to aid the designer in selecting the optimum propeller characteristics for standard type screws. A design method and some diagrams are presented in par. 7-7.5. Thus, in many cases, a suitable screw may be selected with a minimum of computation and no development costs. In addition, the screw may be selected so that it is an off-the-shelf item; hence actual hardware costs are the smallest possible.

Most recent amphibian designs have placed the screw propeller in a tunnel. Fig. 7-66 shows a scale model of the LARC V (note that the propeller here is enclosed in a shroud). The incorporation of a tunnel in the hull is predicated by the requirement for ground clearance when the vehicle is on land. The propeller is well protected from damage from grounding or debris. However, there are disadvantages which must be accepted in such a design. Valuable buoyancy is lost in way of the tunnel and must be recovered by a subsequent increase in draft, beam, or length. The inflow to the propeller is highly disturbed and nonuniform, and the thrust duduction (see par. 7-7.4.4) too may be large. Hence, the propulsive efficiency of this type of propeller-hull combination will suffer. Model test results of the LARC V, Ref. 11, indicate that the propulsive efficiency may be lower than 0.35. In addition, the bare hull drag of the hull is increased by separation in the flow about the tunnel (see par. 7-1.4).

Future developmental work in the form of model tests should reveal areas of improvement in the design of tunnels and propeller-hull-interactions so that the drag and efficiency of this type of configuration would be less objectionable. It is not clear whether the



(A)

(B)

Figure 7-66. Two Views of a Model LARC V

improvements would justify the necessary costs. However, this is an area in which significant gains may be realized.

7-7.3.2 Kort Nozzles

A Kort nozzle propulsor is simply a screw propeller enclosed in a shroud. Nozzles are found on most of the recent amphibian designs including the LARC V and LARC XV. Advantages of nozzles include increased thrust or efficiency, decreased cavitation and vibration, and protection to the propeller. These advantages may not all be achieved simultaneously, however, and the design of a particular nozzle-propeller combination will be directed to exploit one or more of these features.

The increase in thrust or efficiency of a shrouded propeller may be explained as follows: A close-fitting shroud prevents the normal contraction in the slipstream behind the propeller so that the velocity here is less than with an open propeller, hence less kinetic energy is carried off in the propeller race and efficiency is thereby increased. The suppression of the race contraction amounts to a virtual increase in the diameter of the screw. The ideal efficiency of a shrouded propeller can be found if certain assumptions are made about the area of the slipstream far downstream from the propeller disc. Ref. 67 presents a table of gain in ideal efficiency as a function of the loading coefficient where the area of the slipstream is equal to the propeller disc area. These results are reproduced in Table 7-34.

The highest and lowest load coefficients are typical for towboats when towing a barge train or large merchant ships, respectively. An amphibian propeller will normally be moderately loaded with a coefficient somewhere between these two extremes.

In practice, the efficiency gains indicated above will not be attained. First, any added resistance of the shroud should be charged to the propulsion system (actually, a properly designed Kort nozzle will augment the propeller by producing thrust). Second, the propeller diameter must be reduced on the amphibian to fit the shroud and yet maintain the same road clearance. And third, the close proximity to the hull may decrease the shroud's effectiveness. Nevertheless, a Kort nozzle propulsion system should give some improvement in performance over open propellers for amphibians.

Two more desirable qualities of shrouded propellers should be mentioned. First, the propeller naturally operates at higher rotative speeds in a shroud (unless it is designed to prevent cavitation, in which case the nozzle-propeller combination is commonly called a pumpjet). This effect, when combined with the higher speeds of the smaller diameter propeller (dictated by the clearance criteria), will result in weight savings in terms of engines, transmissions, shafting, and in the propeller itself. Second, the bollard pull or towline thrust of a shrouded propeller is higher than for open propellers. Model tests of the utility landing craft, LCU class, (Ref. 68) show that the addition of Kort nozzles increased the bollard pull by about 50 percent. At the same time, however, reverse thrust may be degraded unless

TABLE 7-34 GAIN IN IDEAL EFFICIENCY FOR SHROUDED PROPELLERS

Load Coefficient $CT = \frac{T}{\frac{\rho}{2}\left(V_a^2 A\right)}$	Efficiency Gain in Percent of η_i over unshrouded propeller
0.60	1.0
1.00	2.0
2.00	4.5
4.00	7.8
6.00	10.0
8.00	12.0
10.00	13.7

C_T = loading coefficient, nd
T = thrust, lb
ρ = mass density, slug/ft^2
V_a = advance speed. ft/sec
A = propeller disc area, ft^2
η_i = ideal efficiency defined by momentum theory in par. 7-7.4.1.

special attention is given to this aspect in the shroud design.

Series data for the design of Kort nozzle propulsion systems are not so prevalent as for open screws. Van Manen has published data (Ref. 69), some of which is reproduced in par. 7-7.5. A number of theoretical methods have been developed for shroud design; see Refs. 70 and 71. The design of a Kort nozzle should be only slightly more difficult than an open screw. More data may be available for future nozzle designs.

The disadvantages of a shrouded propeller are generally the same as for open propellers. It is therefore reasonable to suggest that Kort nozzles be considered for all future amphibians with screw propellers in view of the advantages they offer.

7-7.3.3 Controllable-reversible Pitch Propellers

Controllable pitch screws resemble conventional propellers, but mechanical linkages in the hub allow the pitch of the blades to be altered without interrupting rotation of the propeller. Thus the pitch can be adjusted to improve the performance over a wide range of operating conditions such as would be encountered by a tug in the towing and free running conditions. In addition, the blades may be rotated far enough so that the pitch is reversed to give astern thrust without reversing the engines or resorting to reversing transmissions. If twin propellers are installed, steering can be effected by controlling their pitch differentially to vary the thrust between the port and starboard screws. The disadvantages of controllable pitch propellers lie in their increased complexity and accompanying costs. Also, the efficiency at the design point should be slightly lower than for a fixed pitch propeller because a larger hub is required.

It has been suggested that controllable pitch propellers be considered for application on high-speed amphibians where two critical operating points exist. The propeller would normally be designed for the crafts' high-speed operation. But in this case such a propeller may not produce enough thrust at low speed to cross the hump, a characteristic high drag region which planing boats display in transition from the low-speed displacement mode to the high-speed planing mode of operation. The thrust at the hump speed could be increased by about 15 percent for a typical case using a controllable pitch propeller (Ref. 72). This could provide the thrust margin necessary to cross the hump, which seems to be an attractive alternative to either increasing the installed horsepower or designing the propeller with the hump speed as the design point.

Controllable-reversible pitch propellers should be considered for amphibians only if the operating conditions vary widely so that high efficiency is required at more than one operating point, or, if their application will eliminate a transmission which would otherwise be needed for reverse power. Some series test data for open water performance of controllable pitch propellers are presented in Ref. 73.

7-7.3.4 Contra-rotating Propellers

A set of contra-rotating (CR) propellers consists of two axially spaced screws on concentric shafts which rotate in opposite directions. They are more efficient than a single propeller because less energy is lost to rotational flow in the slipstream (rotational velocities are not considered in the simple momentum theory presented in par. 7-7.4.1). Also, for a given disc area the loading on each propeller in a CR set is half of that for a single propeller. CR propellers are widely used on torpedoes because they are

designed to operate with the torque balanced so that there is no rolling moment on the torpedo. CR propellers have also been applied to VTOL aircraft in conjunction with annular wings (Kort nozzles).

The disadvantages of the CR propeller are greater mechanical complexity and more difficult design procedures. There are interactions between the two propellers—the aft propeller operates in the wake of the forward propeller whose inflow is, in turn, affected by the aft propeller. A design method is available, however, which is based on Lerbs' Propeller Theory, Ref. 74.

The contra-rotating propeller may be an attractive propulsor for amphibians where the clearance criteria impose severe limitations on the propeller diameter, with resulting high propeller loading.

7-7.3.5 Vertical Axis Propellers

Unlike the preceding propeller types, vertical axis propellers are completely unrelated to the screw propulsor. These propellers consist of a number of spade-like blades projecting into the water from a circular disc set flush with the bottom of the vessel. The disc rotates about a vertical or near vertical axis. The blades, which are spaced evenly about the periphery of the disc, rotate about their own axes and this rotation is controlled to give thrust force in any desired direction in the plane of the disc. The principles of operation of two vertical axis propellers, the Kirsten-Boeing and Voith-Schneider types, are given in Ref. 67.

Two vertical axis propellers located either fore and aft or port and starboard would provide complete maneuverability in view of their ability to provide thrust and/or moment in any direction. If the blades are permitted to extend below the baseline to get good flow, the vertical axis propellers can achieve high propulsion efficiencies, about as good as for an open screw. Openwater efficiencies as high as η_o = 0.66 have been measured, Ref. 75. However, these propellers would be extremely vulnerable to damage. Large cutouts in the hull are necessary to bring the blades above the baseline which will compromise their performance considerably. Model test results of the research vessel T-AGOR, Ref. 76, show that vertical axis propellers are very sensitive to changes in the hull lines with respect to wake fraction and thrust deduction.

It does not seem likely that vertical axis propellers will be used on amphibians in spite of the advantages they might afford in terms of maneuverability. The large cutouts that would be required for adequate protection would seriously compromise buoyancy and problems with arrangements might be foreseen. In any case, a great deal of model testing would be required to develop a good relationship between the hull and propulsor.

7-7.3.6 Vertical Shaft, Right-angle Drives

Right angle drive units derive their name from the fact that the propeller can be rotated about the vertical shaft axis which is perpendicular to the propeller rotation axis. Thus the propeller thrust may be directed in any direction in the horizontal plane. The propeller may be open or in a Kort nozzle. The propeller may be designed to retract by pivoting the vertical shaft about a horizontal axis. Recent amphibians with vertical shaft outboard drives are the MOFAB and LVW-X3, Refs. 77 and 78. These vehicles are characterized by the absence of any tunnel in the hull. This results in decreased drag and better propulsive efficiency than for comparable amphibians with conventional drives with the propeller in a tunnel.

The main drawback of the vertical shaft, right-angle drive is its extreme complexity and cost. Also, the propeller, which operates below the baseline, is rather vulnerable and must be partially retracted in the surf for protection in which case the performance is seriously compromised.

7-7.3.7 Water Jets

Water jet propulsors consist of an inlet, pump, and jet. Thrust is developed by the change in momentum between water entering the inlet and leaving the jet. The inlet may be a scoop type or flush with the hull. The pump can be centrifugal, mixed flow, or axial-flow depending on the specific pump speed. The jet can be above or below the waterline and can be controllable to provide excellent maneuverability. A most desirable feature of water jets is the fact that all of the machinery is contained within the hull so that shallow water

operations pose no hazard to the propulsor.

Disadvantages of water jets include relatively poor efficiency, high cost, and weight and space penalities. Some of these disadvantages may not be prohibitive if water jets are compared with the alternative of a propeller in a tunnel. The low efficiency of a water jet is acceptable if the hull efficiency and drag of such an amphibian combine so as to require the same shaft horsepower as a hull with a tunnel and its characteristic high drag and low hull efficiency. A large percentage of the weight associated with water jets is due to the water entrained in the system. This may be considered as lost buoyancy and may be no worse than the loss in buoyancy due to a large tunnel cutout in the hull. Similarly, the internal space devoted to the water jet may be of the same order as taken by a large tunnel. Refs. 2, 5, 10, 79, and 80 give data for the design of water jets and their inlets.

7-7.4 Introduction to Theory of Propellers

Before any attempt can be made by the designer to design a propulsor or even select an "off-the-shelf" propeller, he must have an understanding of the basic principles which govern their performance. The presentation which follows is only of an introductory nature. However, it is felt that enough information is contained in this text to allow the designer to select a screw propeller from design charts derived from screw series model tests. If it is decided—in the preliminary design stage—that a detailed design will be made or if another type of propulsor is chosen, the designer is directed to the list of references. In cases where a detailed propeller design will be made, the design charts can still be used to make preliminary propulsion estimates.

7-7.4.1 Momentum Theory

Simple momentum theory was developed by Rankine (1865) and improved by R. E. Froude (1883). It is based on the principle that propellers derive thrust by accelerating the fluid in which they operate. The magnitude of the force is governed by Newton's first law of motion which may be expressed

$$F = ma = m\left(\frac{dv}{dt}\right), \text{ lb} \qquad (7\text{-}71)$$

where

F = force, lb

m = mass, slug

dv/dt = acceleration or time rate of change of the velocity, ft/sec^2

For the special case of steady flow through a propeller this equation may be written

$$T = \dot{m}\ (V_1 - V_a), \text{ lb} \qquad (7\text{-}72)$$

where

T = thrust produced by the propeller, lb

$\dot{m}$ = mass rate of flow through the propeller disc, slug/sec

V_1-V_a = ultimate change in axial velocity of fluid passing through the propeller, where V_a is the initial or upstream velocity and V_1 the final or downstream velocity, ft/sec

We can say then that the thrust is equal to the change in momentum per unit time or the rate change in momentum of the fluid.

The propeller in the simple momentum theory is assumed to be an actuator disc which is capable of imparting an axial change in momentum to the fluid. It is assumed that the velocity imparted to the fluid is uniformly distributed over the disc and that friction is negligible (inviscid flow). The flow is shown in Fig. 7-67 where V_a is the uniform free stream flow equal to the speed of advance of the propeller. V_1 is the velocity in the stream tube far downstream. $V_1 = V_a + u$ so that u is the change in axial velocity imparted by the propeller. The cross-sectional areas of the tube far upstream, at the propeller disc, and far downstream are given by a_1, A, and a_2, respectively.

From the principle of conservation of mass we can write, for the mass rate of flow in through the propeller disc

$$\begin{aligned} m &= \rho V_a a_1 = \rho(V_a + u')A \\ &= \rho(V_a + u)a_2, \text{ slug/sec} \end{aligned} \qquad (7\text{-}73)$$

where

A = disc area, ft^2

ρ = mass density of the fluid, slug/ft^3

(V_a+u') = flow velocity through the plane of the propeller, ft/sec

Eq. 7-72 can be rewritten using the above relation to get

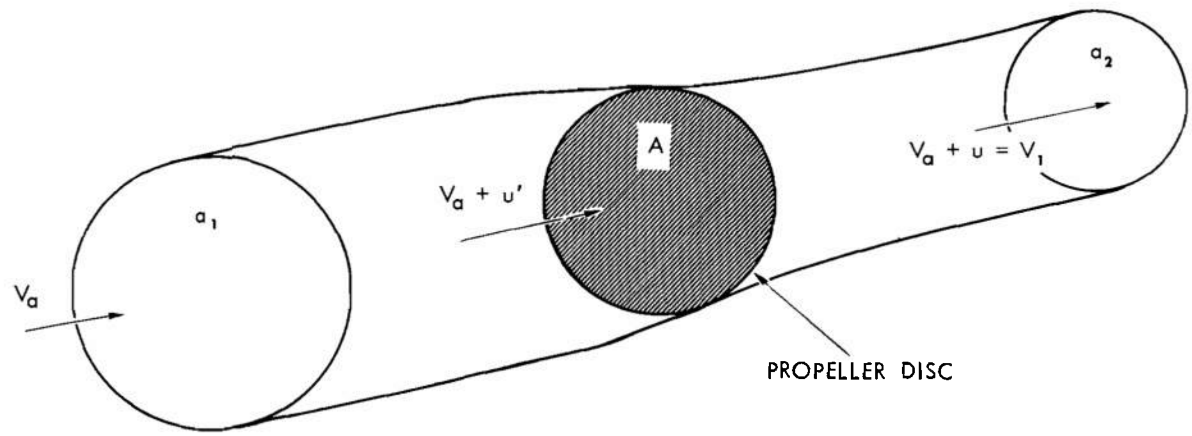


Figure 7-67. Principle of Actuator Disc Propeller

$$T = \rho(V_a + u')A(V_a - V_1) = \rho A(V_a + u')u, \text{ lb} \quad (7\text{-}74)$$

The thrust may also be equated to the change in pressure Δp across the propeller disc times the disc area

$$T = \Delta p A, \text{ lb} \quad (7\text{-}75)$$

The change in pressure is equal to the change in velocity head (dynamic pressure) far upstream and far downstream

$$\Delta p = \tfrac{1}{2}\rho[(V_a + u)^2 - V_a^2] = \rho\left(V_a + \frac{u}{2}\right)u, \text{ lb/ft}^2 \quad (7\text{-}76)$$

Thus,

$$T = \rho A\left(V_a + \frac{u}{2}\right)u, \text{ lb} \quad (7\text{-}77)$$

Comparing Eqs. 7-74 and 7-77, we note that

$$u' = \frac{u}{2} \quad (7\text{-}78)$$

This relation shows that one-half of the ultimate velocity increase imparted to the fluid in the stream tube is imparted ahead of the propeller. Substituting this relation in Eq. 7-73 shows that the area a_2 must be smaller than the disc area A. Experimental evidence confirms this contraction in the propeller race predicted by momentum theory.

Eq. 7-77 may be rewritten in nondimensional form

$$C_T = (2 + s')s' \quad (7\text{-}79)$$

where

load coefficient $C_T = \dfrac{T}{\tfrac{1}{2}\rho V_a^2 A}$, nd

slip ratio $s' = u/V_a$, nd

conversely,

$$s' = \sqrt{C_T + 1} - 1 \quad (7\text{-}80)$$

The efficiency of this ideal actuator disc propelller can be calculated by considering the ratio of the rate of useful work P_{out} to the rate of work done on the fluid P_{in}. The rate of useful work is simply

$$P_{out} = TV_a$$

The power absorbed by the fluid is given by

$$P_{in} = \Delta p\left(V_a + \frac{u}{2}\right)A$$

Thus, the ideal efficiency η_i may be expressed

$$\eta_i = \frac{P_{out}}{P_{in}} = \frac{TV_a}{(V_a + u/2)A\Delta p} = \frac{\Delta p A V_a}{(V_a + u/2)A\Delta p}$$

$$= \frac{V_a}{V_a + u/2} = \frac{2}{2 + s'} = \frac{2}{1 + \sqrt{C_T + 1}} \quad (7\text{-}81)$$

Fig. 7-68 shows a plot of ideal efficiency η_i versus load coefficient C_T.

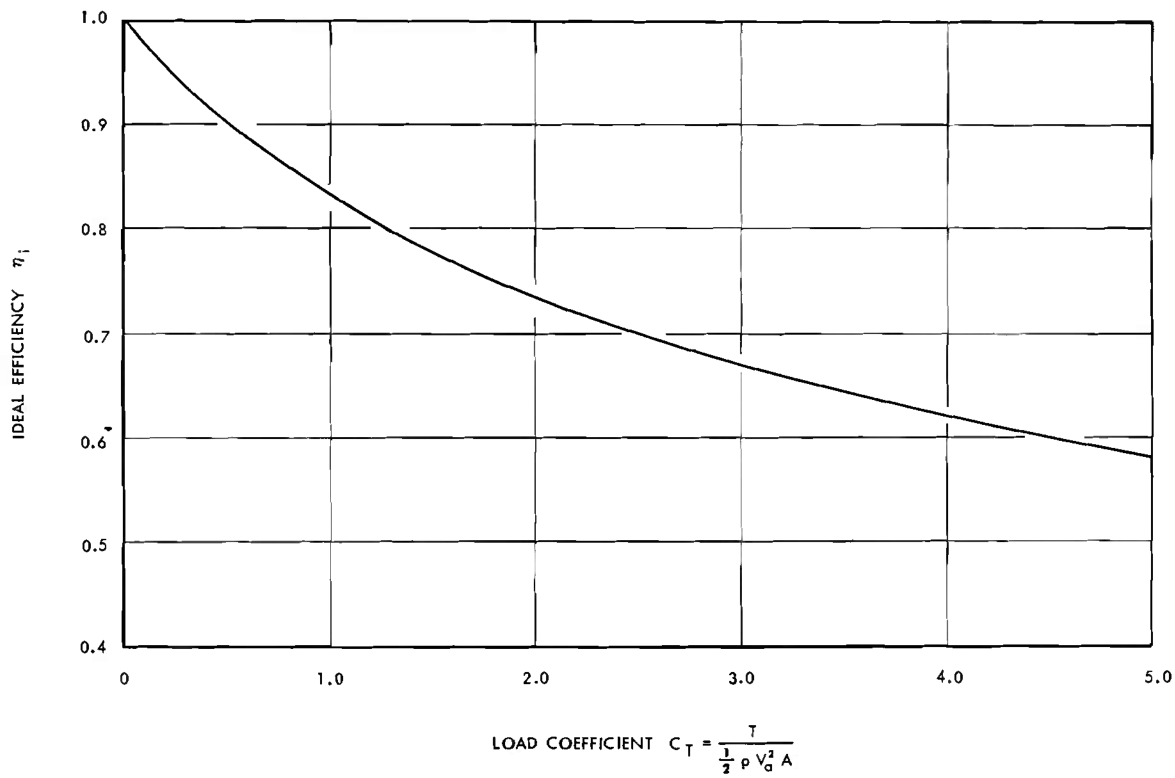


Figure 7-68. Ideal Efficiency (from Simple Momentum Theory)

Some very important conclusions can be drawn from this simple momentum approach to propeller theory. Kinetic energy is carried away in the propeller race which results in ideal efficiencies less than one. The actuator disc is the most ideal propeller imaginable. Hence, the ideal efficiency represents the upper limit of efficiency for real propellers. This upper limit is a function of the propeller loading. A propeller working at a high load coefficient is less efficient (ideally) than one working at a low load coefficient. This leads to the important conclusion that the propeller with the largest disc area is generally most efficient.

Refinements can be made to the momentum theory to account for viscous effects and losses due to rotational velocities in the slipstream. However, the main drawback of any momentum theory is that it yields no information about the geometry (blade shape, pitch distribution, camber, etc.) of the propeller. Theories do exist which take into account the geometrical properties of a screw propeller, and they will be discussed briefly.

7-7.4.2 Blade Element and Circulation Theories

In blade element theory each blade is divided into a number of annular elements. Each element is treated as a foil section. The axial and tangential inflow velocities relative to each blade element are combined vectorially to find the angle of attack (the induced velocities predicted by the momentum theory should be taken into account here). The lift and drag are calculated using suitable airfoil data, and then these forces are resolved into a thrust and torque force. The thrust and torque of the propeller are found by integrating the forces over all of the blade elements.

Blade element theory combined with momentum theory provides a basic conception of screw propeller action, but certain factors are neglected such as interaction between blades and the decrease in lift at the blade tips. These factors are considered in the modern circulation theory. Development of these theories will not be pursued here. However, Refs. 81 through 85 may be studied to gain an understanding of

modern developments in propeller theory.

7-7.4.3 Factors Which Influence Propeller Performance

7-7.4.3.1 Propeller Advance Speed

The performance of a propeller behind a ship differs from that of the same propeller in open water. The flow of fluid into the propeller is affected by the wake of the ship and the flow about the ship is in turn affected by the velocities induced by the propeller. The speed of advance which the propeller "sees" is generally less than the forward speed of the ship. The propeller works in the wake of the vessel which is made up of both potential and boundary layer flow. The relation between ship speed and propeller advance speed is given in terms of the Taylor wake fraction in the equation

$$V_a = V_s(1 - w),\ \text{ft/sec} \tag{7-82}$$

where

V_a = speed of advance of the propeller, ft/sec

V_s = ship speed, ft/sec

w = wake fraction, nd

7-7.4.3.2 Thrust Deduction

The thrust which a propeller must supply is is generally greater than the bare hull towline resistance of a ship. The action of the propeller changes the flow about the hull in its vicinity. The effect is usually to increase the drag on the hull. The relation between towing resistance and necessary thrust for a given speed is given in terms of the thrust deduction t by

$$R_T = T(1 - t),\ \text{lb} \tag{7-83}$$

where

R_T = resistance of the hull alone at some speed V_o, lb

T = thrust which must be supplied by the propeller to sustain the same speed V_o, lb

t = thrust deduction, nd

7-7.4.3.3 Relative Rotative Efficiency

The relative rotative efficiency η_r is used to account for the difference in performance between a propeller operating in a uniform and nonuniform wake. The wake fraction discussed previously is an average figure used to define the average effective speed of advance through the plane of the propeller. However, the wake through the propeller plane is nonuniform and varies both radially and circumferentially. The relative rotative efficiency η_r is defined in the "thrust identity" equations

$$\left.\begin{aligned} Q_o &= Q\eta_r \\ n_o &= n \\ T_o &= T \end{aligned}\right\} \tag{7-84}$$

where

Q = torque, ft-lb

n = rotative speed, rps

T = thrust, lb

and the subscript o denotes uniform flow.

Thus, the relative rotative efficiency η_r relates the efficiency η_B of the propeller in the nonuniform wake of the hull to the open water efficiency η_o of the propeller producing the same thrust at the same rotative speed in uniform flow

$$\eta_B = \frac{TV_s(1 - w)}{2\pi nQ} = \frac{T_o V_a \eta_r}{2\pi n_o Q_o} = \eta_o \eta_r \tag{7-85}$$

7-7.4.3.4 Propulsive Efficiency

The propulsive efficiency η_p is defined as the ratio of effective horsepower *EHP* to delivered horsepower *DHP* (also called the propeller horsepower *PHP*)

$$\eta_p = \frac{EHP}{DHP} = \frac{RV_s}{2\pi Qn} \tag{7-86}$$

This useful factor is related to the open water efficiency η_o, the wake fraction w, the thrust deduction t, and the relative rotative efficiency η_r in the following manner

$$\begin{aligned} \eta_p &= \frac{RV_s}{2\pi Qn} = \frac{T(1 - t)V_s(1 - w)}{2\pi Qn(1 - w)} \\ &= \eta_B \frac{(1 - t)}{(1 - w)} = \eta_o \eta_r \frac{(1 - t)}{(1 - w)} \end{aligned} \tag{7-87}$$

The term $(1 - t)/(1 - w)$ is called the hull

efficiency η_h so that we may write

$$\eta_p = \eta_o \eta_h \eta_r \tag{7-88}$$

Thus, the propulsive efficiency is the product of the open water efficiency, hull efficiency, and relative rotative efficiency. Strictly speaking, the last two factors are not true efficiencies and they may very likely have numerical values greater than unity. Some designers prefer the names hull factor and relative rotative factor for this reason.

7-7.4.3.5 Hull Efficiency

The hull efficiency (hull factor) η_h is increased by reducing the thrust deduction t and increasing the wake fraction w. However, it can be shown that the thrust deduction is related to the wake fraction and, in particular, to the wake from potential flow so that an increase in wake fraction is often accompanied by an increase in the thrust deduction. The hull efficiency can be increased by locating the propeller in a region of high friction wake. The effect in this case is achieved by recovering energy imparted to the fluid due to the drag of the body, and converting it to thrust.

7-7.4.3.6 Propulsion Factor Data

The wake fraction, thrust deduction, and relative rotative efficiency for several amphibians are included in Tables 7-35 through 7-39. These tables give data from model tests. The reference material from which the data were derived is given at the top of each table. All numbers are based on thrust identity which is discussed in detail in par. 7-7.6.3.

7-7.4.4 Propeller Cavitation

7-7.4.4.1 Theory of Cavitation

Cavitation is a type of fluid flow breakdown caused by the formation of cavities in regions of low pressure. Severe cavitation of marine screws should be avoided because it is accompanied by a loss in performance and erosion or pitting of the blade material. The formation of cavities is usually associated with a reduction in local pressure to the vapor pressure, a function of temperature, of the liquid. Depending on the condition of the liquid (air or solid matter content, etc.) cavitation may begin at pressures above or below the vapor pressure. However, for engineering design purposes it may be assumed that inception of cavitation occurs at the vapor pressure p_v. Table 7-40 gives the vapor pressure of water as a function of temperature.

The local pressure about a body in an ideal fluid is related to the local velocity by Bernoulli's theorem which is stated in the equation

$$p = P_o + \frac{\rho}{2}(U^2 - u^2) \tag{7-89}$$

where

p = local pressure, lb/ft^2

P_o = reference pressure, usually taken as the pressure in the undisturbed fluid, lb/ft^2

u = local velocity, ft/sec

U = reference velocity, taken in the undisturbed fluid, ft/sec

ρ = mass density of the fluid, slug/ft^3

Thus, the pressure p may drop to a very low value depending on the velocity u. Cavitation will occur when the local pressure p is less than or equal to the vapor pressure p_v.

The cavitation number σ is defined by the equation

$$\sigma = \frac{P_o - p_v}{\frac{1}{2}(\rho U^2)} = \frac{P_o - p_v}{q} \tag{7-90}$$

where q = dynamic pressure of the undisturbed fluid, lb/ft^2.

The cavitation number is nondimensional and offers a convenient index with which to determine the existence of cavitation and its relative extent. Combining Eqs. 7-89 and 7-90:

$$\sigma = \frac{p - p_v}{q} + \left(\frac{u^2}{U^2} - 1\right) \tag{7-91}$$

Now, we can define the critical cavitation number σ_{cr} as the value at which the local pressure p is equal to the vapor pressure p_v. Thus,

$$\sigma_{cr} = \frac{(u')^2}{U^2} - 1 \tag{7-92}$$

where u' = maximum local velocity of the fluid, ft/sec.

TABLE 7-35 LARC V PROPULSION CHARACTERISTICS

Ref. 11

Ship Particulars:

Length, ft	32.2
Beam, ft	8.7
Trim, ft	0.79 by stern
Appendages	Rudder
Draft, ft	1.56 fwd, 2.35 aft

Propeller Particulars:	Open Propeller (Tests 1 - 4)	Nozzle Propeller (Test 6)
DTMB† propeller number	3660	3760
Diameter, ft	2.5	2.5
Pitch, ft	1.93	2.5
Number of blades	4	3
Expanded area ratio	0.620	0.540
Blade thickness fraction	0.055	0.036
Hub diameter, ft	—	0.38

† David Taylor Model Basin

Ship Speed, mph	7	8	9	7	8	9
EHP, hp	30	50	70	30	50	70
Drag, lb	1600	2340	2910	1600	2340	2910
N, rpm	647	806	911	525	630	705
Thrust, lb	2270	3660	4500	1470	2150	2690
K_T	0.25	0.26	0.25	0.246	0.250	0.250
η_o	0.34	0.31	0.33	0.530	0.525	0.525
$1 - t$	0.73	0.64	0.647	1.09*	1.09*	1.08*
$1 - w$	0.707	0.685	0.748	1.14*	1.175*	1.168*
η_h	1.03	0.96	0.865	0.956*	0.93*	0.93*
η_r	1.0	0.925	0.955	0.96	0.93	0.93
EHP/SHP or η_p	0.35	0.275	0.272	0.485	0.455	0.452

* Note: These include the effects of the nozzle and are not the true propulsion factors; they may be called the apparent propulsion factors.

At cavitation numbers above the critical value there is no cavitation. The degree of cavitation increases with decreasing cavitation numbers below σ_{cr}. The term on the right hand side of Eq. 7-92 may be written as $-\Delta p'/q$ using Bernoulli's equation where $-\Delta p' = P_o - p'$, the maximum local decrease in pressure about a body.

Fig. 7-69 shows the pressure distribution about a typical foil section at some positive angle of attack. In this figure the ordinates of the curves are the differences ($-\Delta p$) between the local pressures and the pressure in the undisturbed medium normalized by the stagnation pressure q; the abscissas are fractional distances along the chord of the foil. The maximum decrease in pressure on the back of this foil occurs at 10 percent of the chord where $-\Delta p'/q = 2.0$. Thus, the critical cavitation number for the foil is $\sigma_{cr} = 2.0$ and inception of

TABLE 7-36 LARC XV PROPULSION CHARACTERISTICS

Ref. 57

Ship Particulars:

Length, ft	41.8
Beam, ft	12.5
Draft, ft	2.47 fwd, 2.90 aft
Appendages	Rudder and Kort nozzle

Propeller Particulars:

DTMB† Propeller No.	3760
Diameter, ft	3.0
Pitch, ft	3.0
No. of blades	3
Expanded area ratio	0.54
Blade thickness fraction	0.036
Hub diameter, ft	0.46

† David Taylor Model Basin

Ship Speed, mph	8	10	12
Thrust, lb	3,600	6,500	10,900
SHP, hp	190	450	765
N, rpm	580	765	980
K_T	0.242	0.250	0.256
η_o	0.540	0.525	0.513
$1-w$	1.345*	1.370*	1.425*

*Note: This wake fraction includes the effect of the nozzle on the inflow to the screw, hence, it is not a true wake fraction.

TABLE 7-37 DRAKE PROPULSION CHARACTERISTICS

Ref. 54

Vehicle displacement, lb	45,000
Propeller diameter, in.	31 (Twin screws in tunnels)
Propeller pitch, in	24.5

Ship Speed, mph	7	8	9
SHP, hp	38	68	111
N, rpm	494	595	706
$1-t$	0.84	0.815	0.78
$1-w$	0.82	0.860	0.85
η_h	1.03	0.950	0.92

cavitation would occur on the back of the foil at the 10 percent chord point. For a numerical example, assume that this foil (σ_{cr} = 2.0) is operating in fresh water at a depth such that the undisturbed pressure P_o = 20 psia and the temperature is 59°F (at 59°F, from Table 7-40, p_v = 0.025 psia)

$$\sigma_{cr} = \frac{P_o - p_v}{\frac{1}{2}(\rho U_{cr}^2)}$$

$$U_{cr}^2 = \frac{P_o - p_v}{\frac{1}{2}(\rho\sigma_{cr})} = \frac{(20 - 0.25)144}{(0.5)1.938(2.0)} = \frac{(19.75)144}{1.938}$$

$$U_{cr} = 38.3 \text{ ft/sec}$$

TABLE 7-38 GULL PROPULSION CHARACTERISTICS

Ref. 92

Ship Particulars:		Propeller Particulars:	
Length, ft	35	Diameter, ft	3.01
Beam, ft	9.17	Pitch (0.7*R*), ft	2.93
Displacement, lb	44,703	Number of blades	3

Test - 4M Single Screw in Tunnel

Ship Speed, mph	7	8	9
EHP, hp	42	66	91
$1 - w$	1.04	1.02	1.06
$1 - t$	1.11	1.03	0.98
η_h	1.07	1.01	0.93

TABLE 7-39 LVW (HULL MODE) PROPULSION CHARACTERISTICS

Ref. 39

Hull Particulars:		Propeller Particulars:	
Length, ft	3.60	Diameter, ft	2.0 (Twin screws)
Beam, ft	11.33	Pitch, ft	2.4
Displacement, lb	38,000	No. of blades	3
Trim, ft	1.5 by stern	Expanded area ratio	0.945
		Blade thickness fraction	0.06

	Wheel Flaps Up		Wheel Flaps Down	
Ship Speed, mph	6.93	11.5	6.93	11.5
$1 - w$	0.92	1.15	1.04	1.10
$1 - t$	0.70	0.80	0.92	0.71
η_h	0.77	0.70	0.89	0.65
η_o	0.35	0.40	0.51	0.42
η_r	0.80	1.01	0.79	0.89
η_p	0.22	0.28	0.36	0.24

The foil will be free of cavitation below the critical speed. At any other angle of attack the pressure distribution will be changed and, thus, the critical speed will be different.

A schematic diagram of the critical cavitation number for cavitation inception on a foil as a function of angle of attack is shown in Fig. 7-70. The shaded side of each curve is the region in which each type of cavity will occur—leading edge face, leading edge back, and back bubble. Another type of cavitation common to propellers, tip vortex cavitation, is not shown here. This type of diagram is often referred to as a cavitation bucket.

The lift coefficient of the foil shown in Fig. 7-69 is proportional to the area enclosed by the

TABLE 7-40 VAPOR PRESSURE OF FRESH WATER*

Temperature T		Vapor Pressure	
°F	°C	p_v, psia	h_v, ft of water
32	0	0.09	0.20
40	4.4	0.12	0.28
50	10.0	0.18	0.41
59	15.0	0.25	0.57
60	15.6	0.26	0.59
70	21.1	0.36	0.84
80	26.7	0.51	1.17
90	32.2	0.70	1.61
100	37.8	0.95	2.19
120	48.9	1.7	3.9
150	65.6	3.7	8.6
180	82.2	7.5	17.3
212	100.0	14.7	33.9

*Salt water has a vapor pressure about 98 percent of that of fresh water.

pressure curve. Note that the area about the back of the foil is greater than the area about the face. The largest part of the lift (thrust for a propeller) is contributed by the low pressure region on the back of the foil. The critical cavitation number is a function of the peak of the pressure curve and not the area (lift) enclosed by the curve. Hence, in designing a propeller to avoid cavitation it is desirable to use sections whose pressure distributions are nearly rectangular and are devoid of sharp peaks.

Many alternative forms of the basic cavitation number are used when dealing with propellers. However, all of these are easily related to the number in which the reference pressure P_o is the static pressure at the axis of the screw and the dynamic pressure q is related to the speed of advance of the screw. The general cavitation number σ_o for propellers is given by

$$\sigma_o = \frac{P_o - p_v}{½(\rho V_a^2)} = \frac{(h_A + h_H) - h_v}{\dfrac{V_a^2}{2g}} \tag{7-93}$$

where

h_A = atmospheric pressure, ft(of propeller water)

h_H = depth of immersion of the screw axis, ft

h_v = vapor pressure (Table 7-40), ft

V_a = advance speed, ft/sec

g = acceleration, constant, ft/sec^2

7-7.4.4.2 Cavitation Effects

The damage to screw propellers caused by cavitation is well documented. The erosion of blade material is associated with the collapse of cavities as they flow from a low pressure region to a region where the local pressure is greater than the cavity vapor pressure. The mechanism of this damage will not be pursued here.

The effect of cavitation on performance can be seen in Fig. 7-71 which shows the thrust coefficient K_T and torque coefficient K_Q as functions of speed coefficient J with cavitation number σ_o as a parameter. These coefficients are defined in the figure. The lower the cavitation number, the greater the loss in thrust and torque (larger cavities). The loss in torque is generally proportional to the loss in thrust so that efficiency is relatively independent of the cavitation number. The loss in thrust is caused by a change in the pressure distribution over the backs of the blades. The minimum pressure which the liquid can sustain is its vapor pressure.

While the curve of efficiency is not necessarily affected by cavitation, a propeller on a ship will operate at lower than design efficiency if it

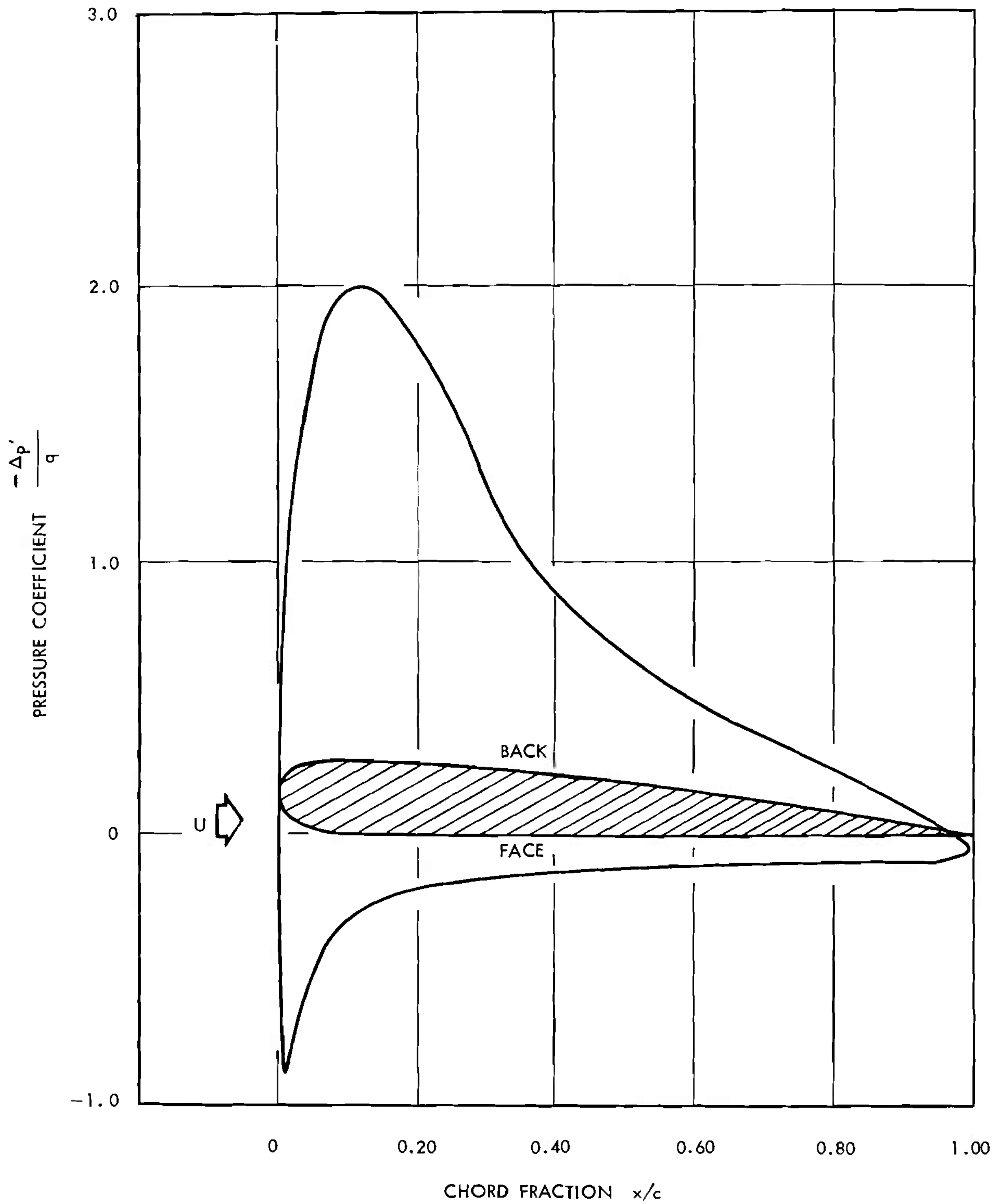


Figure 7-69. Pressure Distribution for a Typical Foil Section

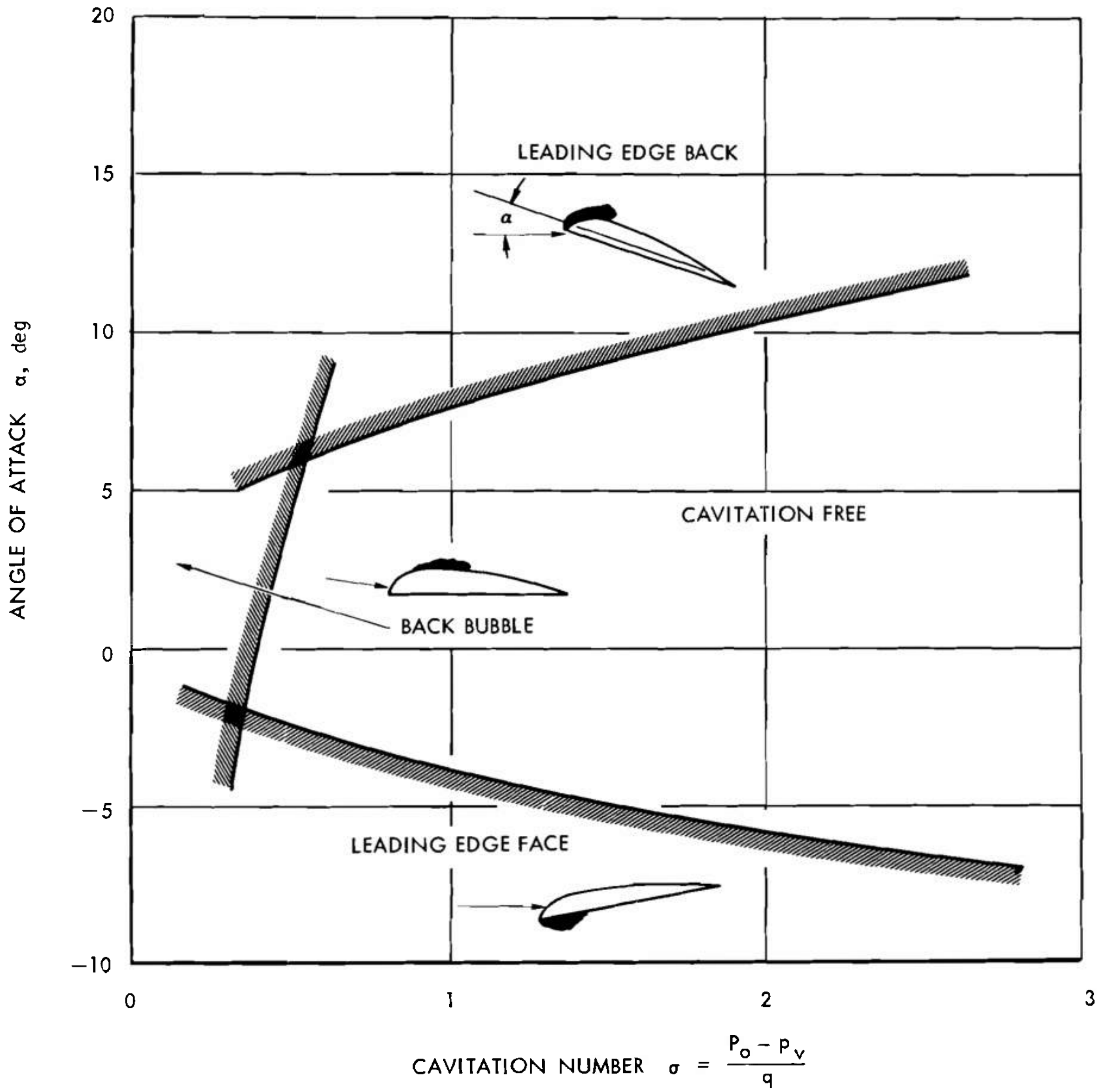


Figure 7-70. Schematic Diagram of Critical Cavitation Number as a Function of Foil Angle of Attack

should cavitate heavily. This is due to the fact that it must operate at higher rpm to develop the design thrust, hence it operates at lower speed coefficient J where the efficiency η is lower (see Fig. 7-95).

Prediction of cavitation for propellers and design features to prevent cavitation are discussed in par. 7-7.5.3. Ref. 87 contains information on the mechanism of cavitation and includes a list of 101 references.

7-7.5 SCREW PROPELLER DESIGN

Screws can be selected by the use of design charts based on standard series data or designed by the application of detailed calculation methods. Chart methods enable standard-type screws to be designed with a minimum of computation. The geometrical features of such screws generally lead to simplification of manufacture. Detailed calculation methods

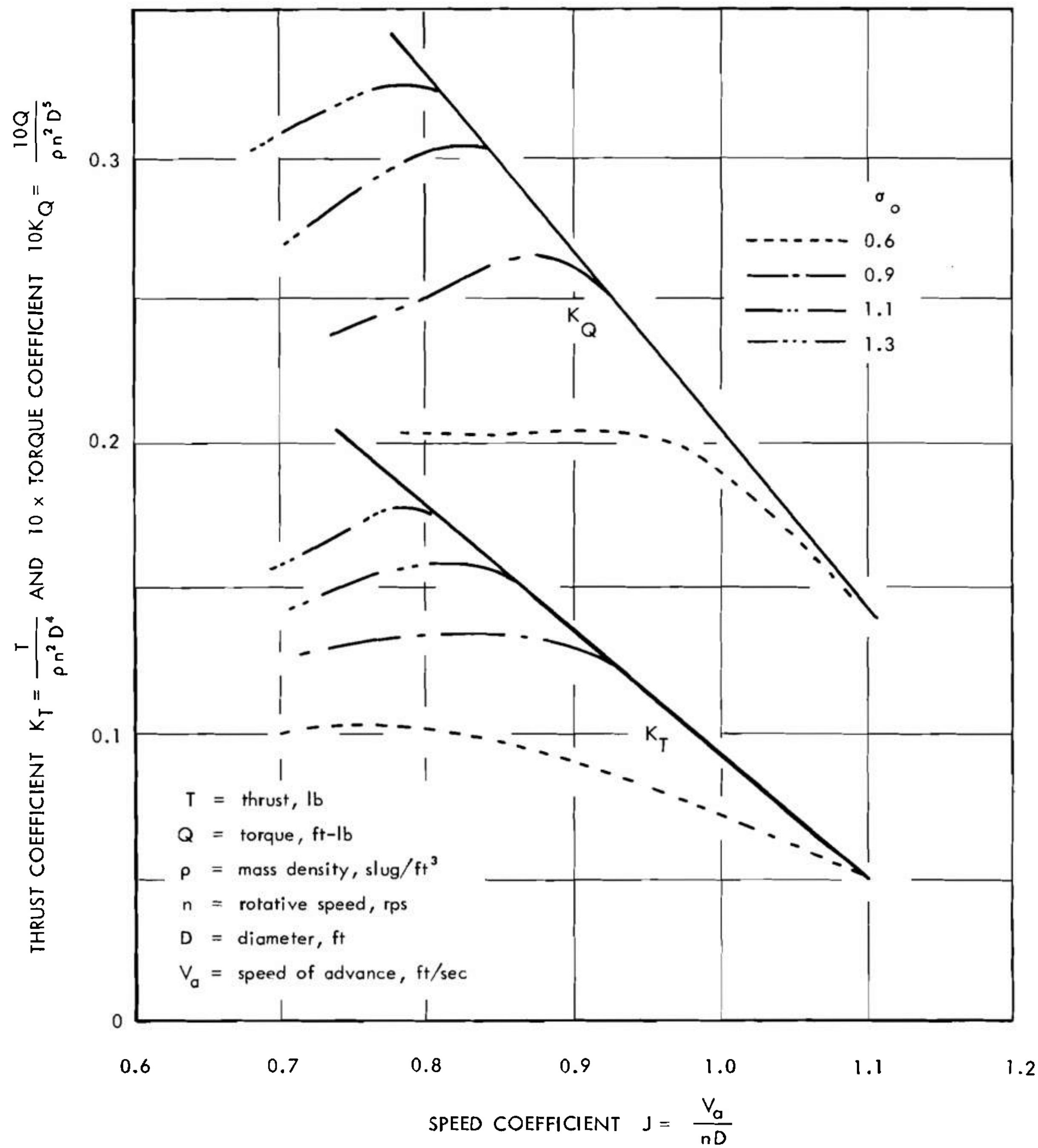


Figure 7-71. Typical Propeller Performance Curves as a Function of Cavitation Number σ_o

involve more computation and result in more complex characteristics which can mean that the screw may operate more efficiently and with a greater margin against cavitation than would be possible for a standard-type screw. For screws required to operate at only light-duty low-speed conditions in relatively uniform flow, there is little to be gained in adopting nonstandard type screws which require detailed calculations. This is not true regarding the design of heavily loaded and wake-adapted propellers.

The combination of chart and detailed design methods provides a procedure which includes some of the advantages of both and may be profitably applied to the design of amphibian propellers. Much of the preliminary design work can be based on standard series design charts. In the early stages of the design many of the requirements will be provisional and some parameters such as hull resistance, wake fraction, and thrust deduction will be based on engineering estimates. The final design work can be related to detailed calculation methods once the requirements have been firmly established and model tests have been performed to confirm the choice of hull and determine its characteristics.

Design charts based on the Troost B-Series of propellers are included in this text to provide a convenient source for amphibian screw propeller design. A limited number of charts are also presented for nozzle-propeller combinations based on data from Ref. 69. Numerical examples are also included to illustrate the use of these charts.

Space limitations preclude the development of detailed design data and methods in this text. However, Refs. 81 to 86 should be studied if a detailed propeller design is to be undertaken.

7-7.5.1 Requirements

The design of a propeller starts with the preliminary propulsion estimate. The designer is given the required design speed, estimates of EHP, thrust deduction, wake fraction, and design stipulations (if any) such as maximum diameter, minimum rpm, type and number of blades, and specified material. With this information, the designer can use design charts to estimate the propeller characteristics, propeller efficiency, propulsive efficiency, and engine horsepower. In the final stages of the design, after a particular power plant has been chosen and model tests have been made, the designer will select the optimum screw characteristics from design charts or from detailed design calculations.

The designer will supply data on the screw particulars—e.g., type of screw, number of blades, diameter, blade area, pitch, rake, skew, thickness, blade section offsets and outlines, etc.—including drawings where necessary. The maximum speed and efficiency should be estimated along with performance over a range of operating conditions at various vehicle displacements and in rough water.

7-7.5.2 Selecting Blade Characteristics

Before we describe the use of screw-series data diagrams in selecting the optimum propeller characteristics, some general discussion on the influence that various factors have on performance is in order.

7-7.5.2.1 Design Variables

a. *Number of Blades Z.* The number of blades is usually selected to minimize vibrations. Variations in wake around the propeller plane induce fluctuations in thrust and torque on each blade. The number of blades determines the phase relationships between blades, and thus whether the oscillating forces tend to augment or cancel one another. Also, the blade frequency (nZ) must not coincide with the natural and lower harmonic frequencies of the hull or machinery. Propeller efficiency is usually a secondary consideration in the choice of the number of blades.

b. *Expanded Area Ratio EAR.* EAR is the ratio of the expanded area (see par. 7-7.5.9) of all the blades to disc area. The efficiency of the screw decreases with increasing blade area due to increasing surface friction. In nearly all cases, however, the minimum allowable blade area is determined by the requirement that the propeller shall be noncavitating, as far as possible. If there is little risk of cavitation, the minimum blade area is restricted by considerations of backing power and strength to about $EAR = 0.35$. Below this value the necessary short, thick sections will result in decreased efficiencies.

c. *Diameter D.* The diameter of a propeller

for an amphibian is usually limited by practical considerations, i.e., tip clearance. In general, the largest allowable propeller will be the most efficient, as shown by momentum theory, par. 7-7.4.1. If diameter is unrestricted, the optimum diameter can be found from design charts and will depend on the restrictions placed on rpm.

d. *Rate of Rotation rpm.* The propeller rpm is often limited by the engine characteristics and reduction ratio of the transmission system. If the optimum rpm for a specified diameter lies below the allowable range, the pitch must be reduced below optimum to increase rpm with some loss in efficiency.

e. *Pitch Ratio P/D.* The pitch ratio is simply the ratio of pitch to diameter. The pitch is the length that the propeller would move in the axial direction per revolution with zero slip. The pitch determines the rate at which a propeller must rotate to produce a given loading coefficient C_T. The pitch ratio for best efficiency is low ($P/D \approx 0.6$) for heavily loaded propellers, and high ($P/D \approx 1.4$) for lightly loaded propellers. Many propellers are designed with radial variation in pitch. This is usually in the form of a pitch reduction at the blade roots and blade tips. Moderate variations in pitch distribution have little, if any, effect on the overall efficiency of a screw but can be favorable from a cavitation standpoint.

f. *Blade Section Shape.* The blade section shapes have important effects on efficiency and the cavitation properties of the screw. Comparison of lift-drag indicates that airfoil shapes are preferable for the inner portions of the blade but are equal to ogival sections for the outer portions of the blade from an efficiency standpoint. However, ogival sections are preferable for the outer portions of the blade from the cavitation standpoint. The Troost Series B screws employ airfoil inner sections and ogival outer sections as shown in Fig. 7-72.

g. *Blade Outline.* The blade outline for best efficiency is one for which the enclosed area distribution approaches that of an ellipse with the major axis along a radial. Blades with broader tips absorb more power and give more thrust at the same slip but are generally less efficient, except for shrouded propellers.

h. *Thickness.* The thickness of the blade sections is determined by strength considerations. The distribution of thickness between the root and tip may be specified as linear or may be determined by strength calculations at a number of radii in order to save weight through the most efficient use of material. Increased thickness results in lower efficiencies and increased cavitation.

i. *Rake and Skew.* Rake and skew are defined by Fig. 7-72. Forward rake is rarely found since it serves no useful purpose. Rake aft is commonly found and is used to increase the tip clearance from the hull. Rake is limited for heavily loaded propellers by the centrifugal bending stresses which are induced. Skew is used to smooth out the fluctuations in thrust caused by abrupt changes in wake behind sternposts and struts.

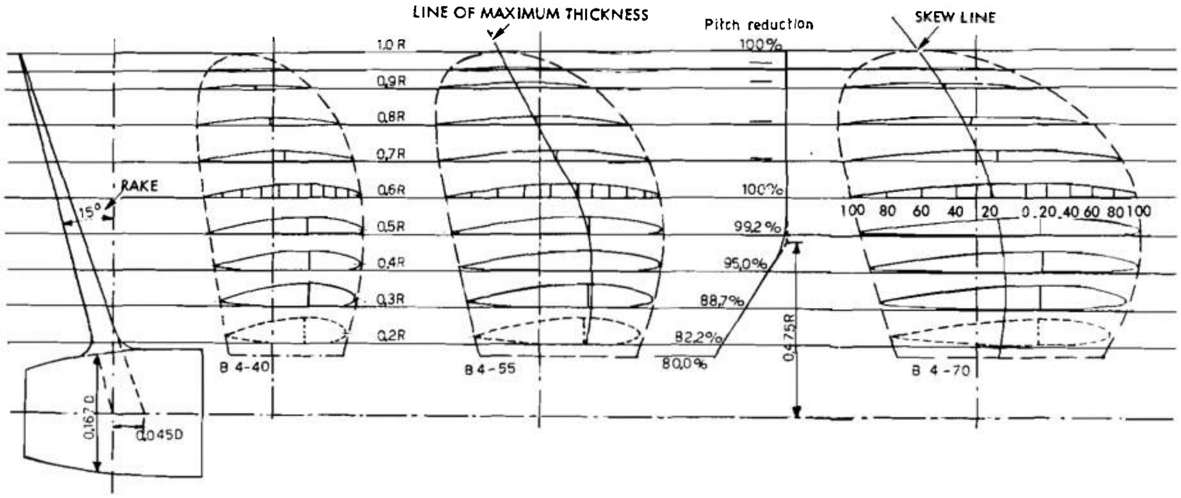


Figure 7-72. General Arrangement of Troost B-4 Series Screws

7-7.5.2.2 Design Charts

The K_T-J diagrams for the Troost B Standard Screw Series are presented in Figs. 7-73 to 7-81. Table 7-41 summarizes the main characteristics of the B-series.

The contour, ordinates, and thickness of the blades are presented in Table 7-42. All have 15 deg rake and some skew. The skew and maximum thickness lines for the 2- and 3-bladed screws differ from that for the 4- and 5-bladed screws. The maximum thickness varies linearly from root to tip. The pitch is defined as the pitch of the blade face rather than the usual nose-tail line. The pitch is constant except for the 4-bladed screws which have a 20 percent reduction at the root. Fig. 7-72 shows the expanded outlines of the 4-bladed series.

Figs. 7-82 to 7-85 are K_T-J diagrams for several Troost B-Series propellers in a shroud. The nozzle characteristics are defined in Fig. 7-86 and are tabulated on each K_T-J diagram.

7-7.5.2.3 Procedures

The use of these design charts for three design problems will be discussed briefly. Numerical examples are included in par. 7-7.5.8 to illustrate these procedures.

Problem No. 1

Given: Design speed V_s, towing resistance R, diameter D, thrust deduction t, wake fraction w, and relative rotative efficiency η_r.

Problem: To find optimum rpm, pitch ratio P/D, and necessary delivered power *DHP*.

Analysis: a. The coefficient K_{T_d} is calculated from the definition

$$K_{T_d} = \frac{K_T}{J^2} = \frac{T}{\rho V_a^2 D^2} = \frac{R_T}{\rho(1-t)(1-w)^2 V_s^2 D^2} \tag{7-94}$$

where units are consistent so that K_{T_d} is dimensionless.

With the calculated value of K_{T_d} a parabola from the equation $K_T = J^2 K_{T_d}$ is drawn on the appropriate K_T-J diagram for the desired number of blades and area ratio. The point where this parabola crosses the curve "η_{max} for K_{T_d} constant" is the design point. K_T and J are read off directly. η_o and P/D are found by linear interpolation between lines of constant efficiency and pitch ratio.

b. Rotative speed rps is calculated from the speed coefficient J:

$$n = \frac{V_s(1-w)}{JD}, \text{ rps} \tag{7-95}$$

c. *DHP* is found from

$$DHP = \frac{R_T(1-w)V_s}{(1-t)\eta_r\eta_o 550}, \text{ hp} \tag{7-96}$$

Problem No. 2

Given: The same as the previous problem with the exception that n is given.

TABLE 7-41 CHARACTERISTICS OF TROOST B-SERIES PROPELLERS

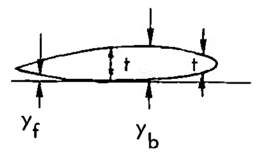
Distribution	Number of Blades Z	Expanded Area Ratio *EAR*	Blade Thickness Fraction *BTF*	Pitch Distribution
B-2.30	2	0.30		Constant
B-3.35	3	0.35	0.50	Constant
B-3.40	3	0.50	0.50	Constant
B-3.55	3	0.60	0.50	Constant
B-4.40	4	0.40	0.45	20%
B-4.55	4	0.55	0.45	Decrease
B-4.70	4	0.70	0.45	at hub
B-5.45	5	0.45	0.40	Constant
B-5.60	5	0.60	0.40	Constant

TABLE 7-42 TROOST B-SERIES PROFILES, ORDINATES, AND THICKNESS

NOTES:

1. r/R RADIUS FRACTION: THE SECTION RADIUS DIVIDED BY THE PROPELLER RADIUS.
2. t AND y_f ARE GIVEN AS A PERCENTAGE OF THE MAXIMUM THICKNESS AT THE CORRESPONDING SECTION
3. T.E. AND L.E. REFER TO TRAILING AND LEADING EDGE, RESPECTIVELY

DISTANCE OF THE ORDINATES FROM THE MAXIMUM THICKNESSES



$$y_b = y_f + t$$

PERCENTAGE OF LENGTH FROM POINT OF MAXIMUM THICKNESS TO TRAILING EDGE | PERCENTAGE OF LENGTH FROM POINT OF MAXIMUM THICKNESS TO LEADING EDGE

r/R	100%	80%	60%	40%	20%	0%	20%	40%	60%	80%	90%	100%	r/R
ORDINATES FOR FACE = y_f													
0.6	5.10										0.50	10.25	0.6
0.5	9.70	1.75							0.35	1.70	4.45	17.05	0.5
0.4	17.85	6.20	1.50					0.30	1.75	5.90	9.90	24.35	0.4
0.3	25.35	12.20	5.80	1.70			0.45	1.30	4.65	10.90	16.25	31.00	0.3
0.2	30.00	18.20	10.90	5.45	1.55		0.45	2.80	7.40	15.50	21.65	36.75	0.2
B 2 - 3 - 4 - 5 - 6 - 7 THICKNESS = t													
0.95		44.80	72.00	88.80	97.20	1.00	97.20	88.80	72.00	44.80	29.50		0.95
0.90		45.15	70.00	87.00	97.00	1.00	97.00	87.00	70.00	45.15	30.10		0.90
0.80		40.95	67.80	85.30	96.70	1.00	97.00	86.30	70.00	47.00	31.65		0.80
0.70		39.40	66.90	84.90	96.65	1.00	97.60	89.15	73.20	49.00	32.95		0.70
0.60		40.20	67.15	85.40	96.80	1.00	98.10	90.65	75.40	51.65	34.35		0.60
0.50		41.65	68.40	86.10	96.95	1.00	97.00	89.70	77.40	55.75	37.55		0.50
0.40		41.50	68.75	86.55	97.00	1.00	97.50	90.10	77.20	55.70	38.85		0.40
0.30		38.75	65.80	85.10	96.80	1.00	97.70	90.05	75.80	53.95	37.45		0.30
0.20		35.15	61.75	81.45	94.90	1.00	97.70	89.65	74.95	51.95	35.55		0.20

THE PERCENTAGES OF THE ORDINATES AND THICKNESSES HAVE REFERENCE TO THE MAXIMUM THICKNESS OF THE CORRESPONDING SECTIONS

BLADE CONTOUR

THE FOLLOWING VALUES ARE PERCENTAGES OF THE BLADE WIDTH AT 0.6 R

B 4 - B 5 - B 6 - B 7					B 2 - B 3					
r/R	T.E.	L.E.	TOTAL WIDTH	POINT OF MAXIMUM THICKNESS TO GENERATOR LINE	r/R	T.E.	L.E.	TOTAL WIDTH	POINT OF MAXIMUM THICKNESS TO GENERATOR LINE	r/R
1.00	20.14	—	—	-20.14	1.00	14.87	—	—	-14.87	1.00
0.95	42.71	11.75	54.46	-15.48	0.95	39.86	18.10	57.96	-10.88	0.95
0.90	47.00	25.35	72.35	-10.83	0.90	45.01	30.76	75.77	- 7.12	0.90
0.80	48.35	41.65	90.00	- 1.37	0.80	47.77	45.08	92.85	+ 0.70	0.80
0.70	46.68	51.40	98.08	+ 8.05	0.70	46.95	52.24	99.19	+ 8.40	0.70
0.60	43.92	56.08	100.00	+17.18	0.60	44.08	55.92	100.00	+16.22	0.60
0.50	40.78	57.60	98.38	+22.68	0.50	40.53	56.52	97.05	+22.07	0.50
0.40	37.30	56.32	93.62	+23.65	0.40	36.62	54.91	91.53	+22.97	0.40
0.30	33.32	52.64	85.96	+22.55	0.30	32.67	51.24	83.91	+21.87	0.30
0.20	29.18	46.90	76.08	+20.27	0.20	28.68	46.05	74.73	+19.89	0.20

BLADE WIDTH AT 0.6 R

2 BLADE = 1.1040 x EAR x D

3 BLADE = 0.7396 x EAR x D

4 BLADE = 0.5467 x EAR x D

5 BLADE = 0.4373 x EAR x D

6 BLADE = 0.3628 x EAR x D

7 BLADE = 0.3174 x EAR x D

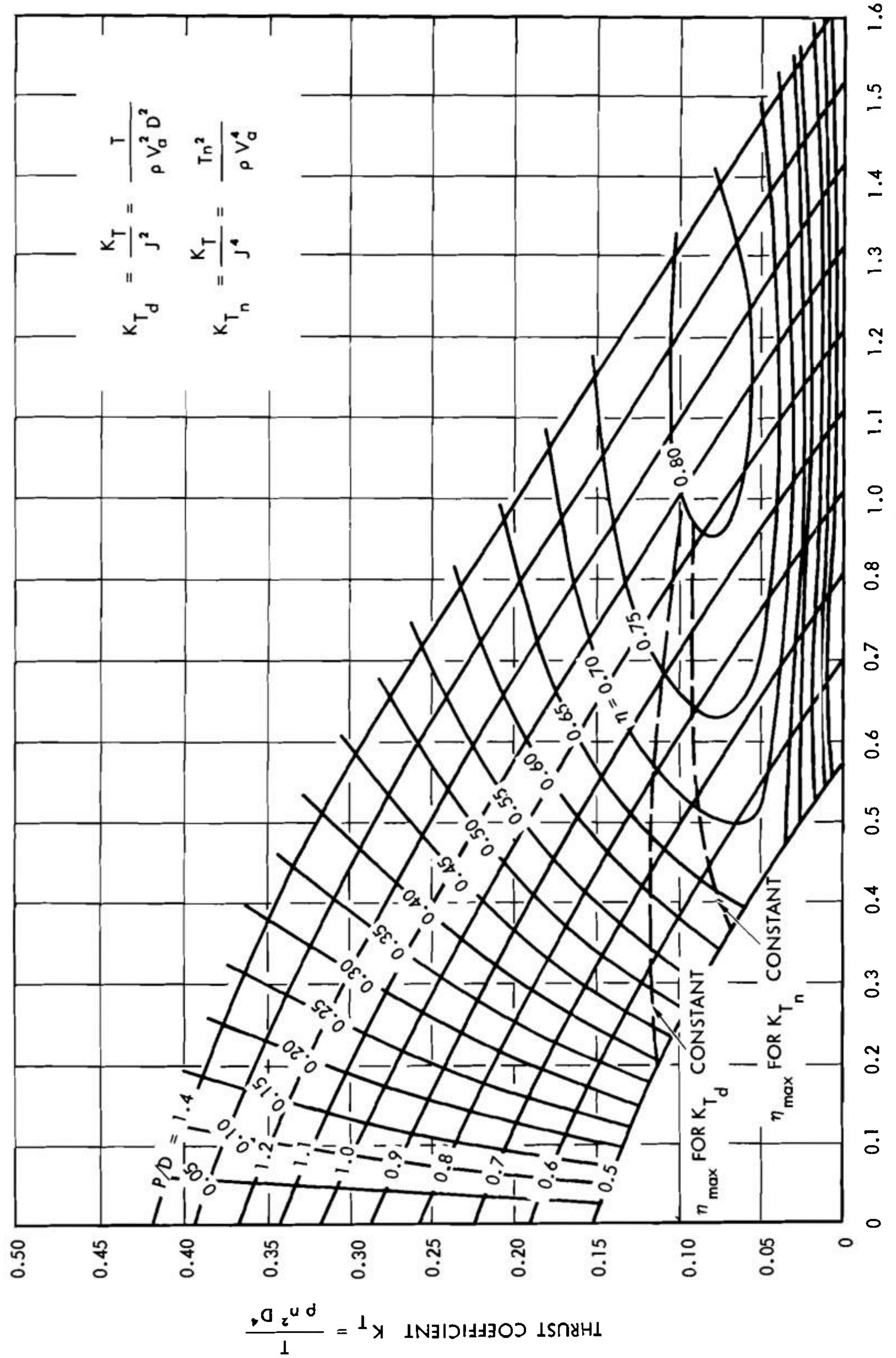


Figure 7-73. Design Chart Based on Troost B-2.30 Series Data

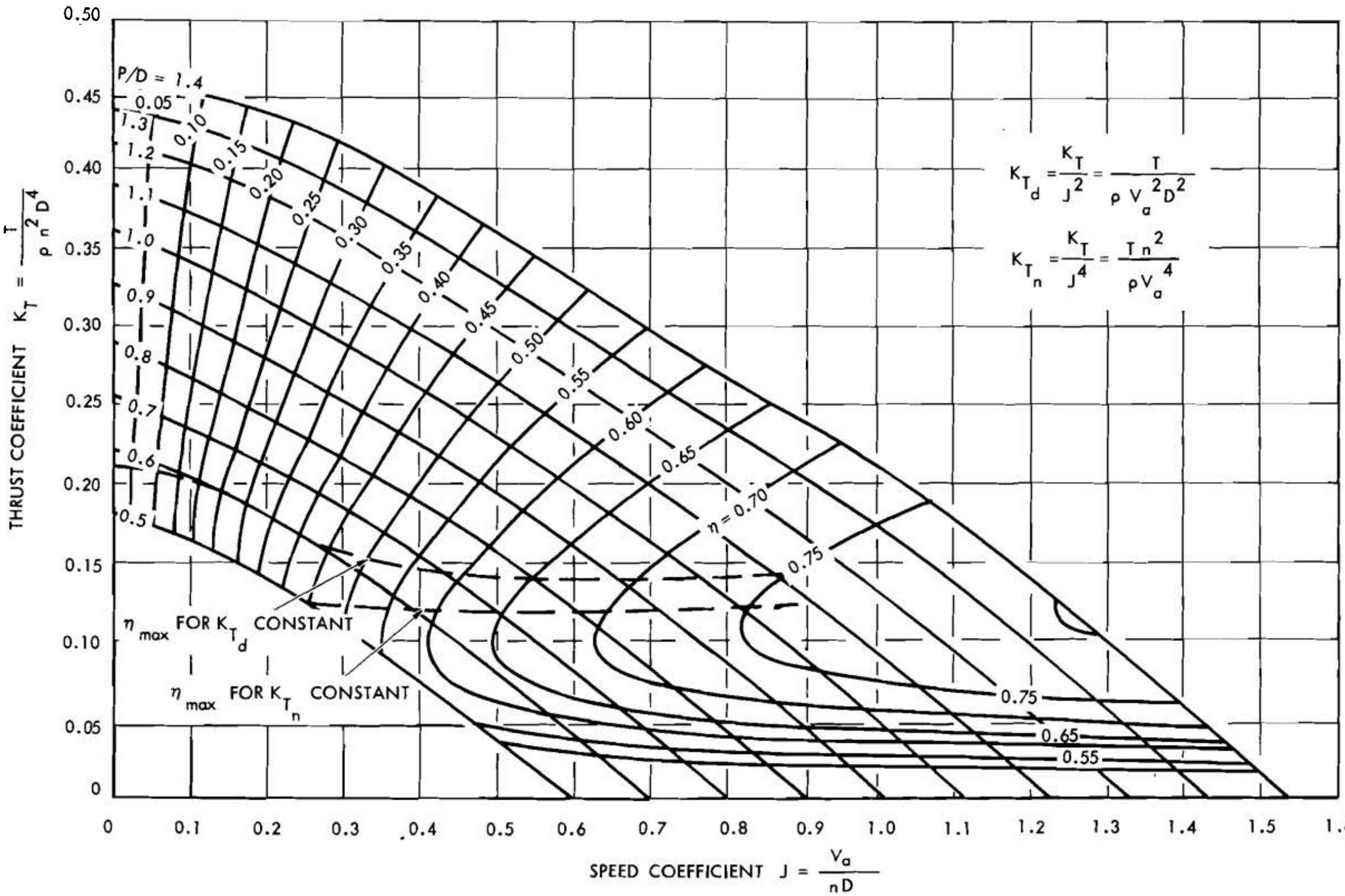


Figure 7-74. Design Chart Based on Troost B-3.35 Series Data

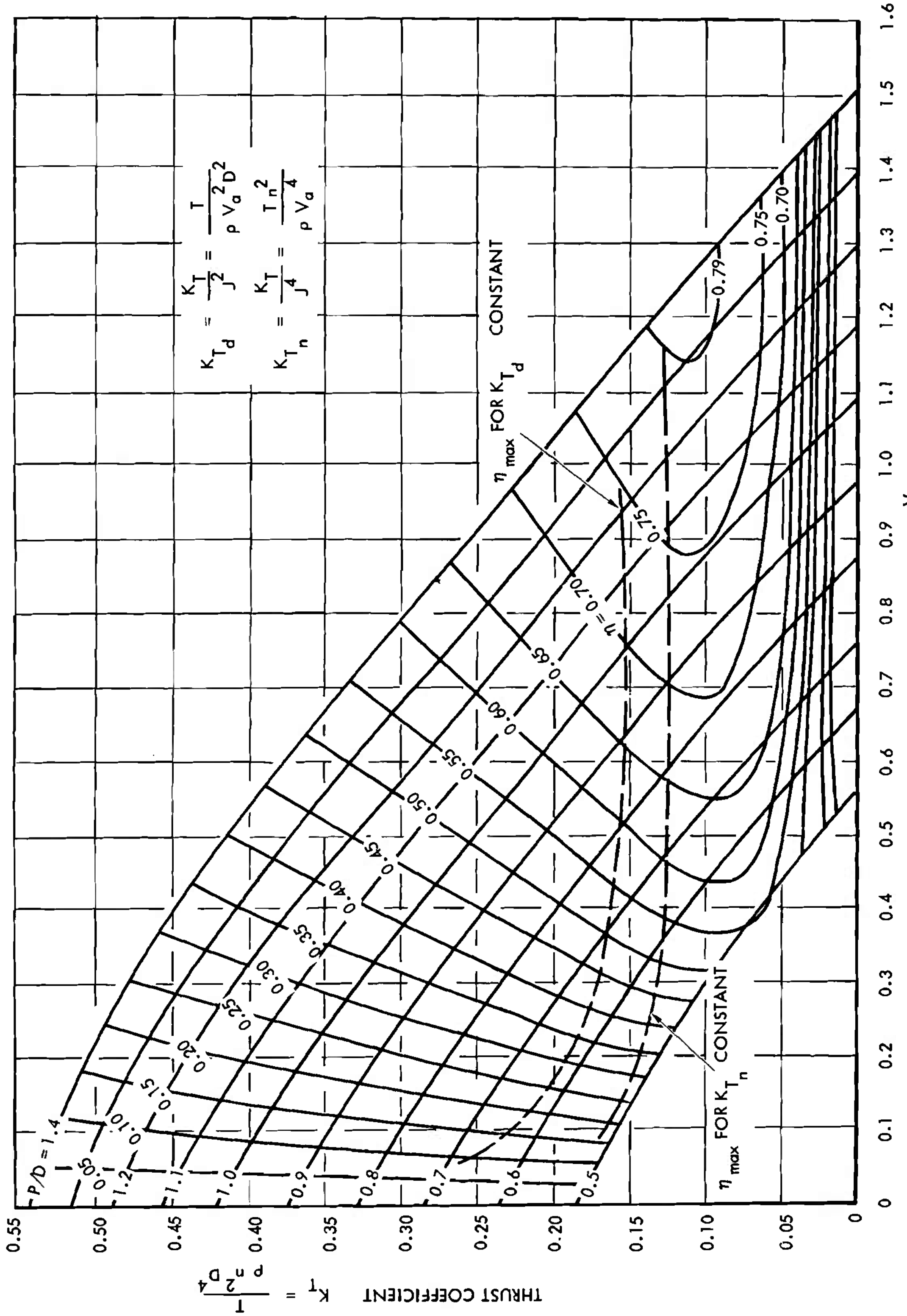


Figure 7-75. Design Chart Based on Troost B-3.50 Series Data

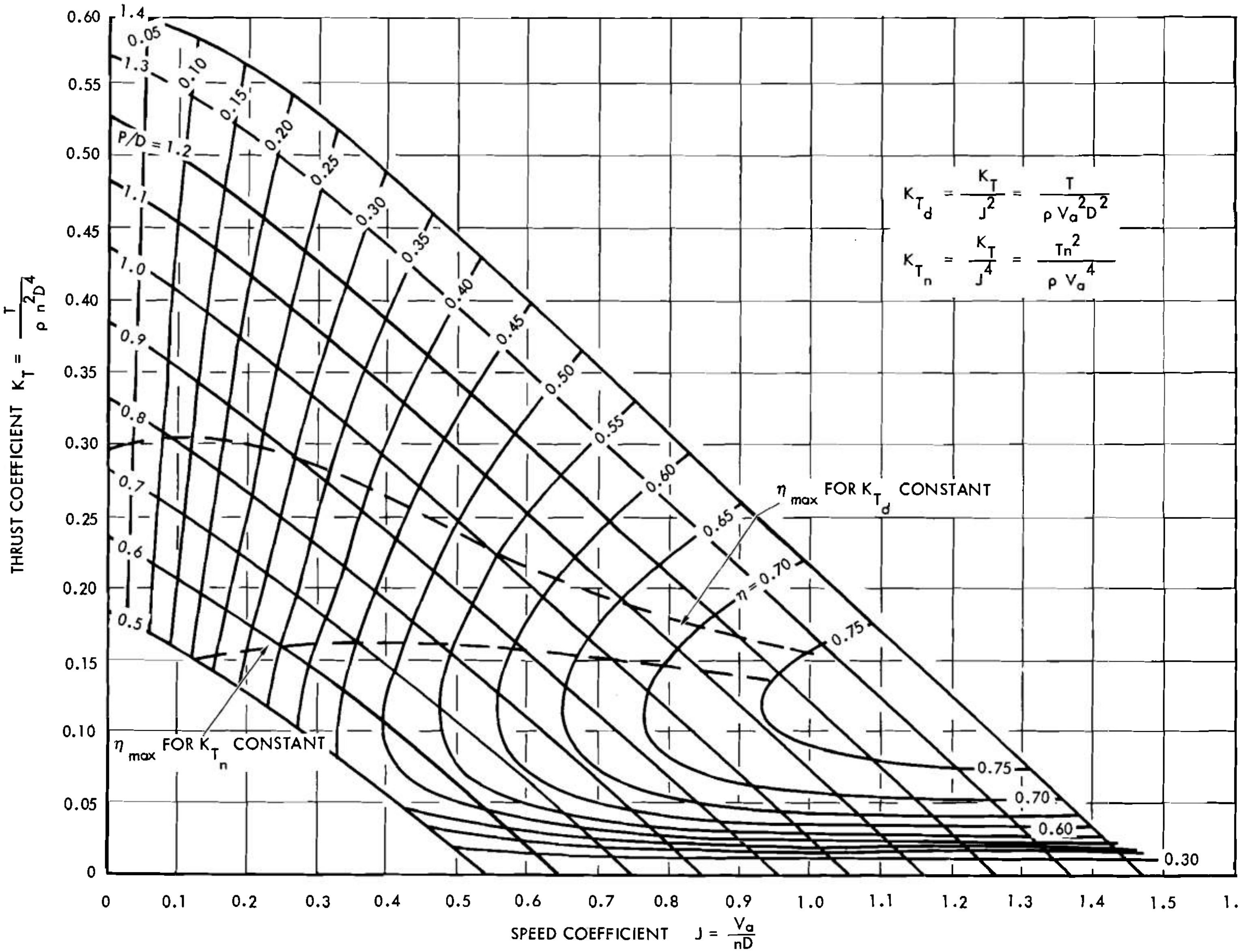


Figure 7-76. Design Chart Based on Troost B-3.65 Series Data

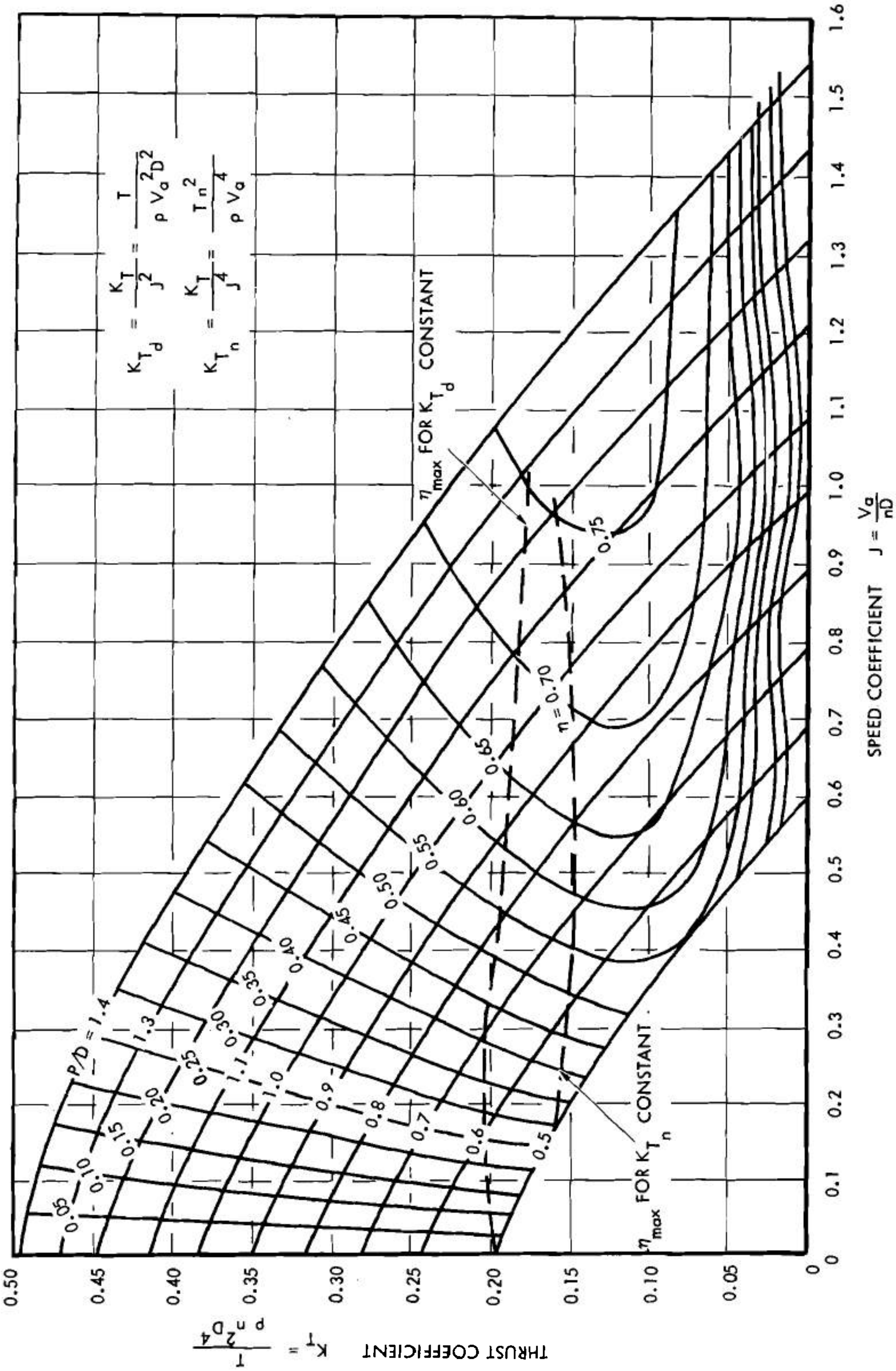


Figure 7-77. Design Chart Based on Troost B-4.40 Series Data

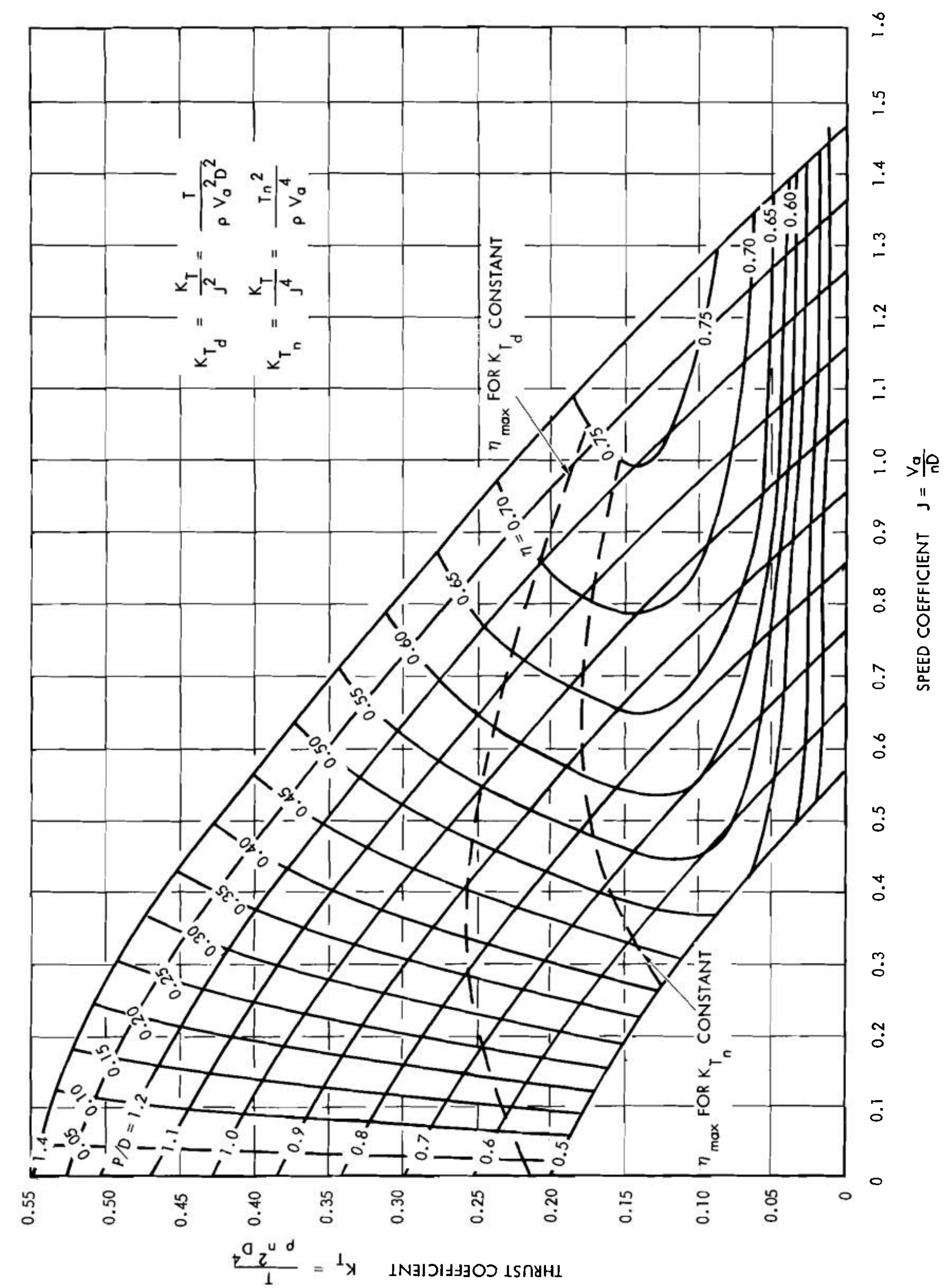


Figure 7-78. Design Chart Based on Troost B-4.55 Series Data

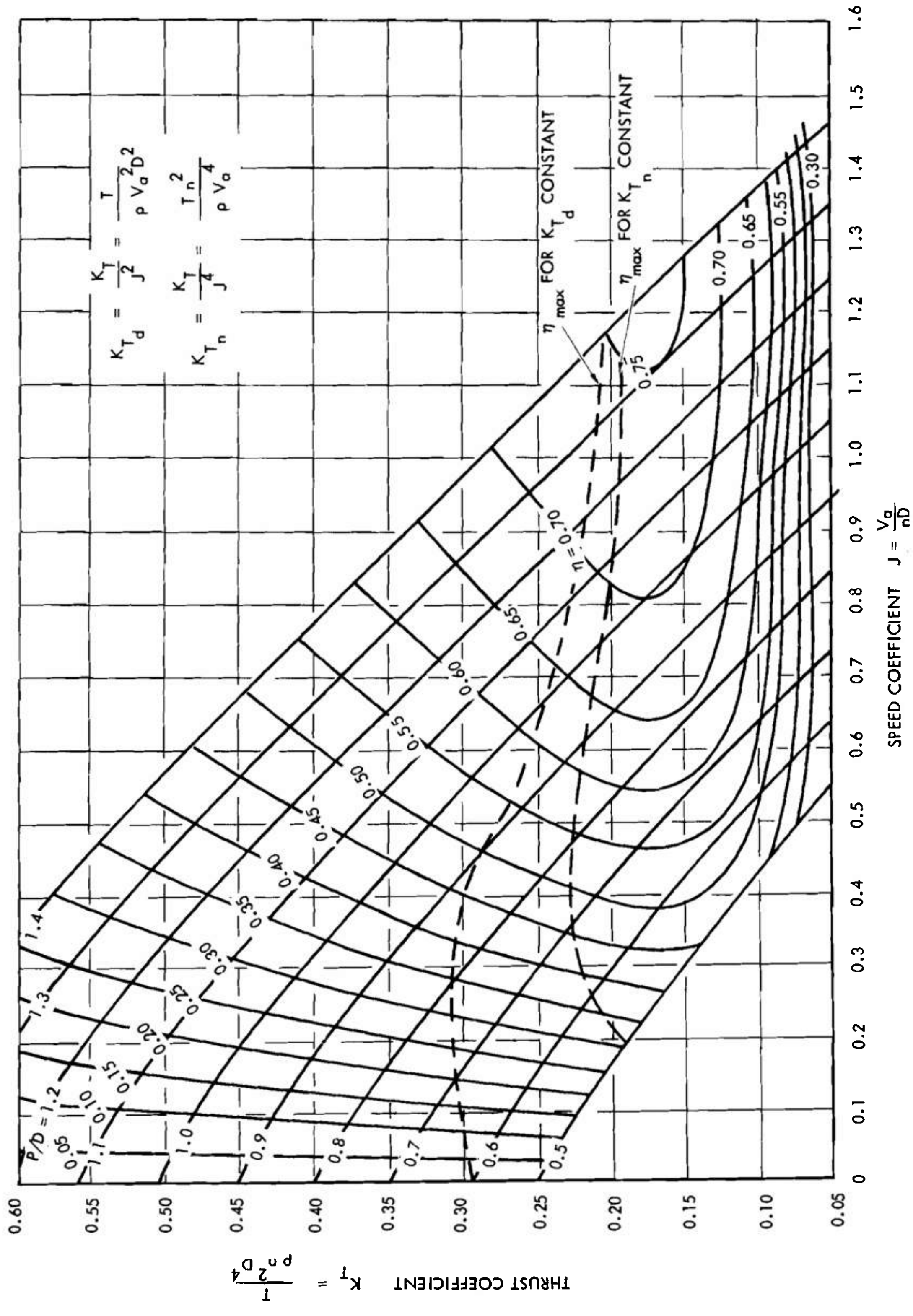


Figure 7-79. Design Chart Based on Troost B-4.70 Series Data

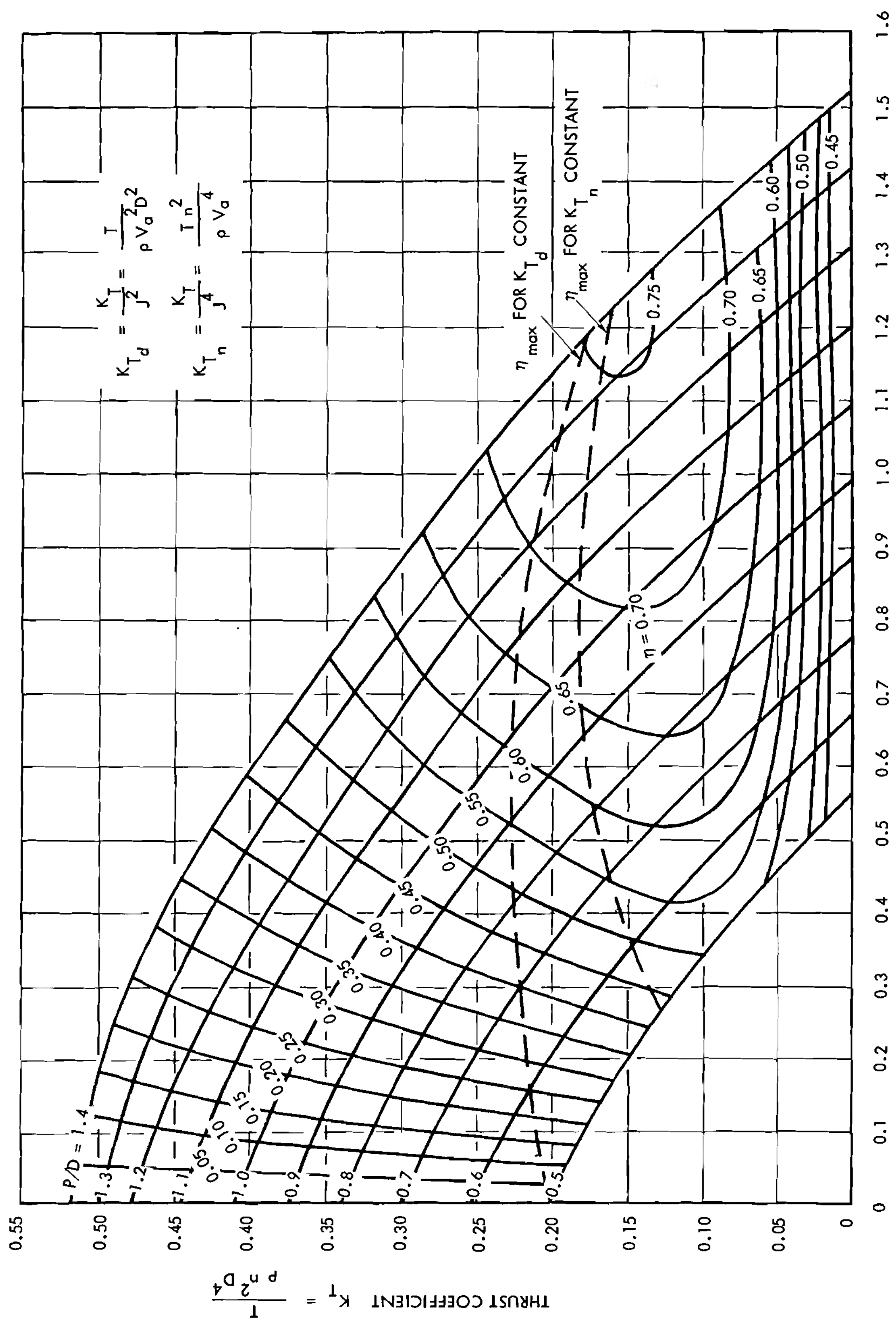


Figure 7-80. Design Chart Based on Troost B-5.45 Series Data

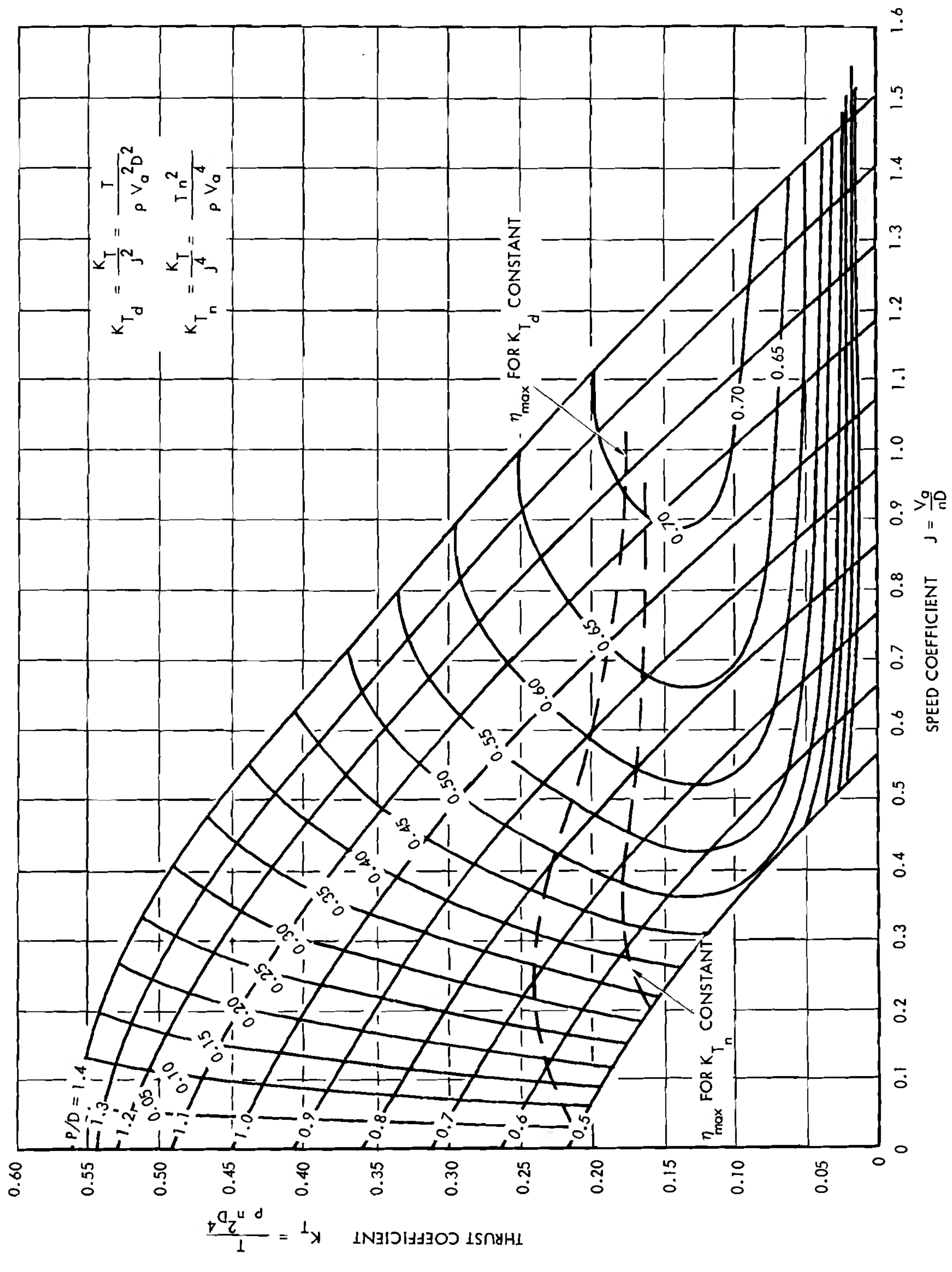


Figure 7-81. Design Chart Based on Troost B-5.60 Series Data

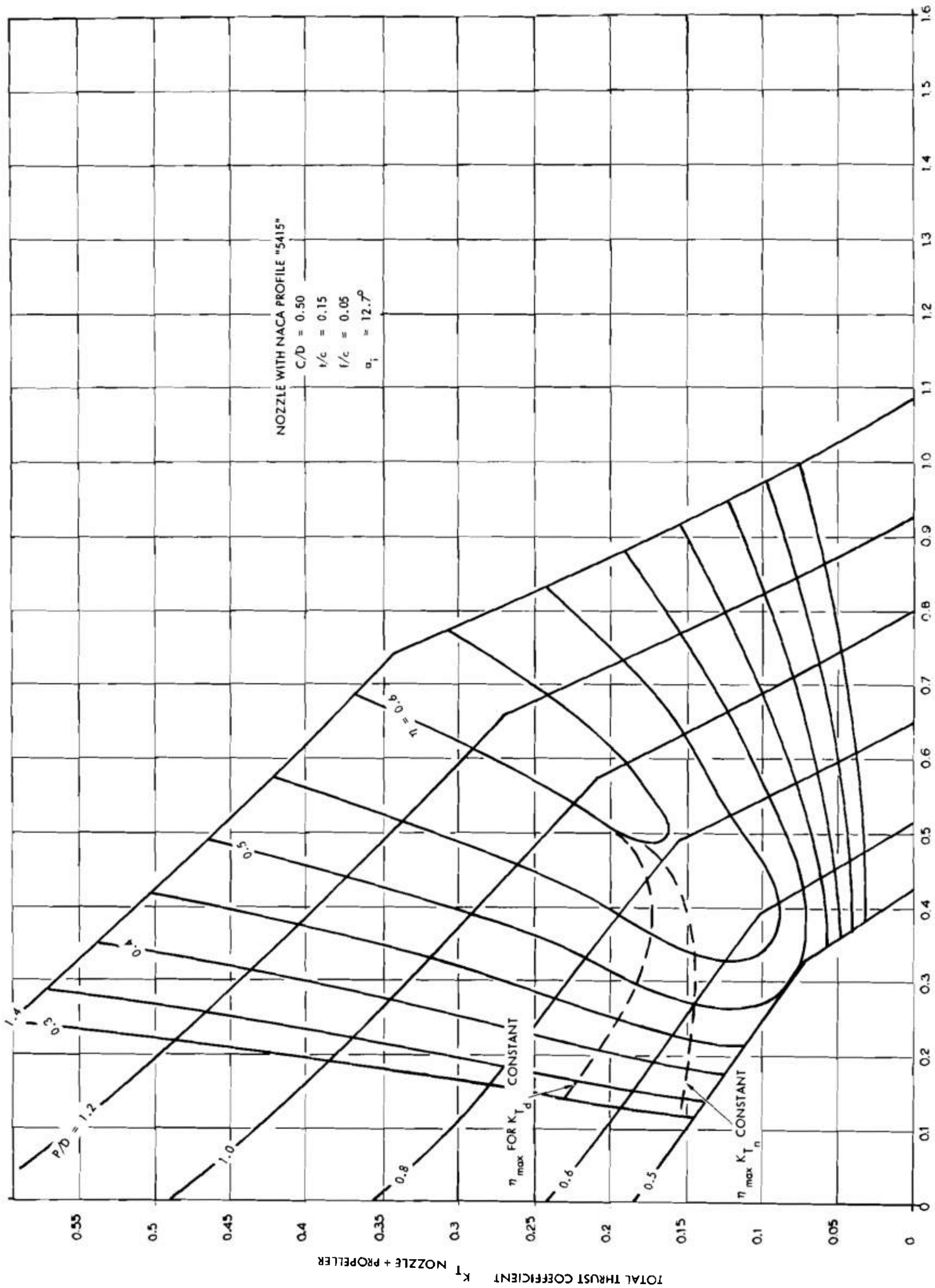


Figure 7-82. Design Chart for Troost B-3.50 Screw in Nozzle

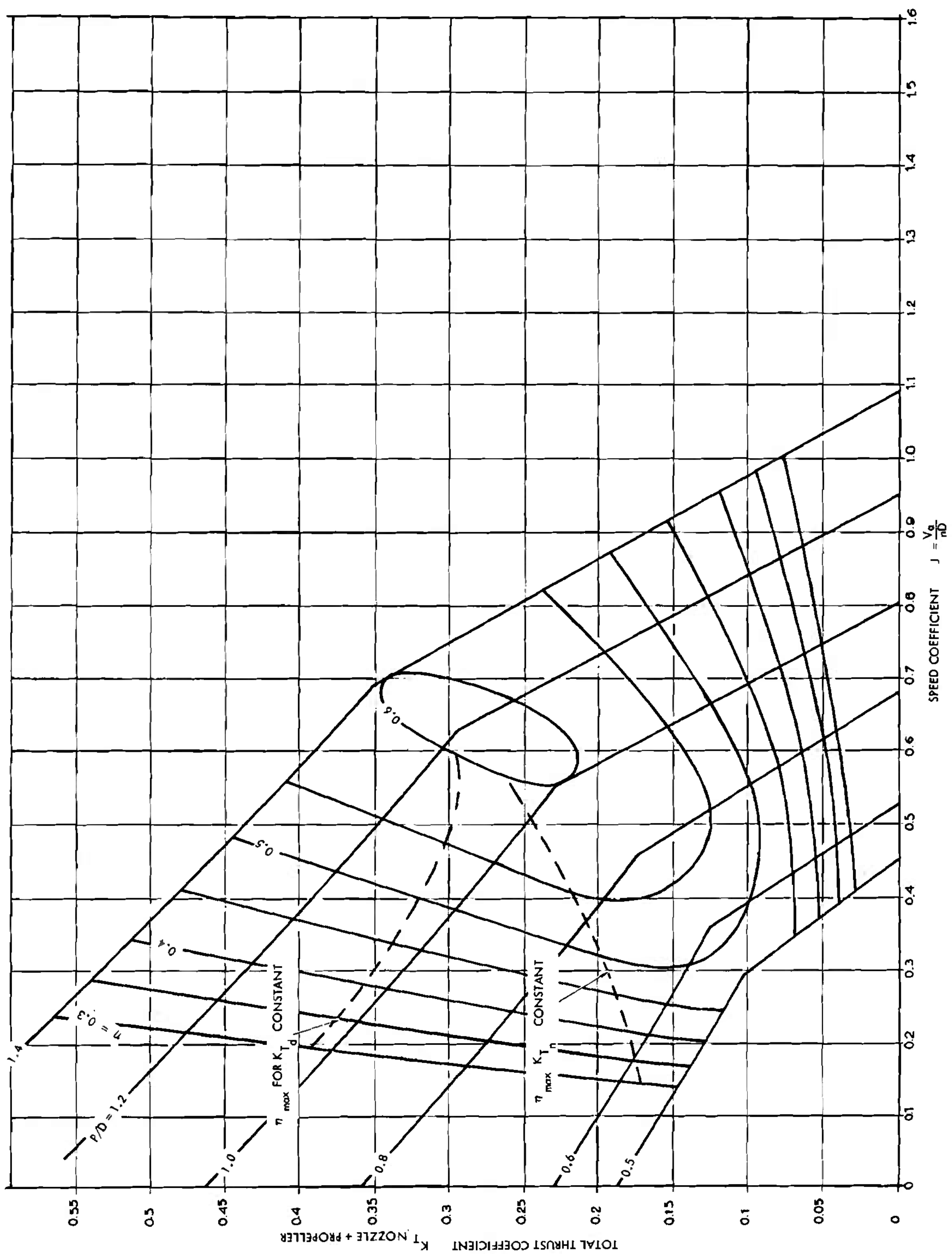


Figure 7-83. Design Chart for Troost B-4.40 Screw in Nozzle

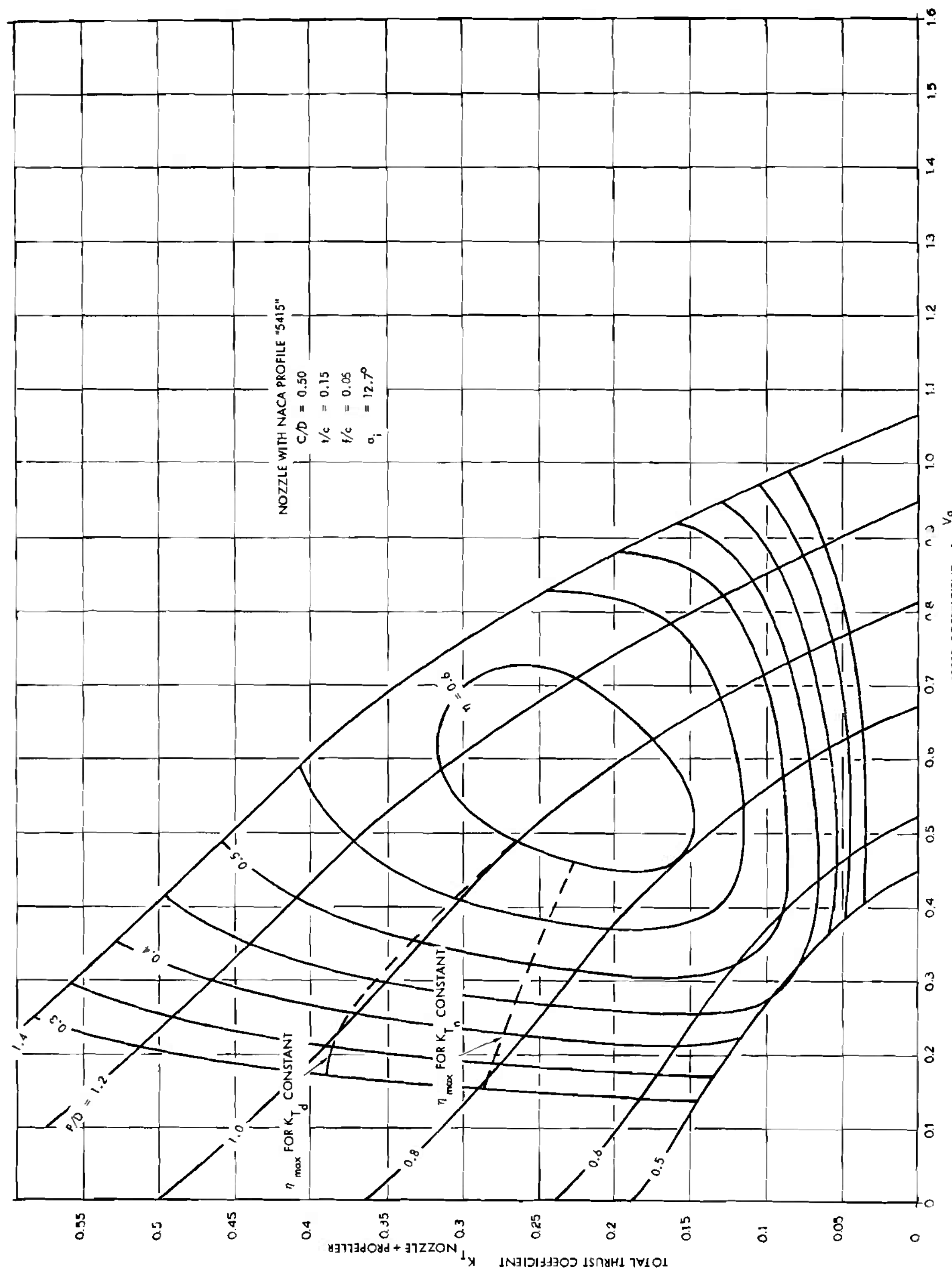


Figure 7-84. Design Chart for Troost B-4.55 Screw in Nozzle

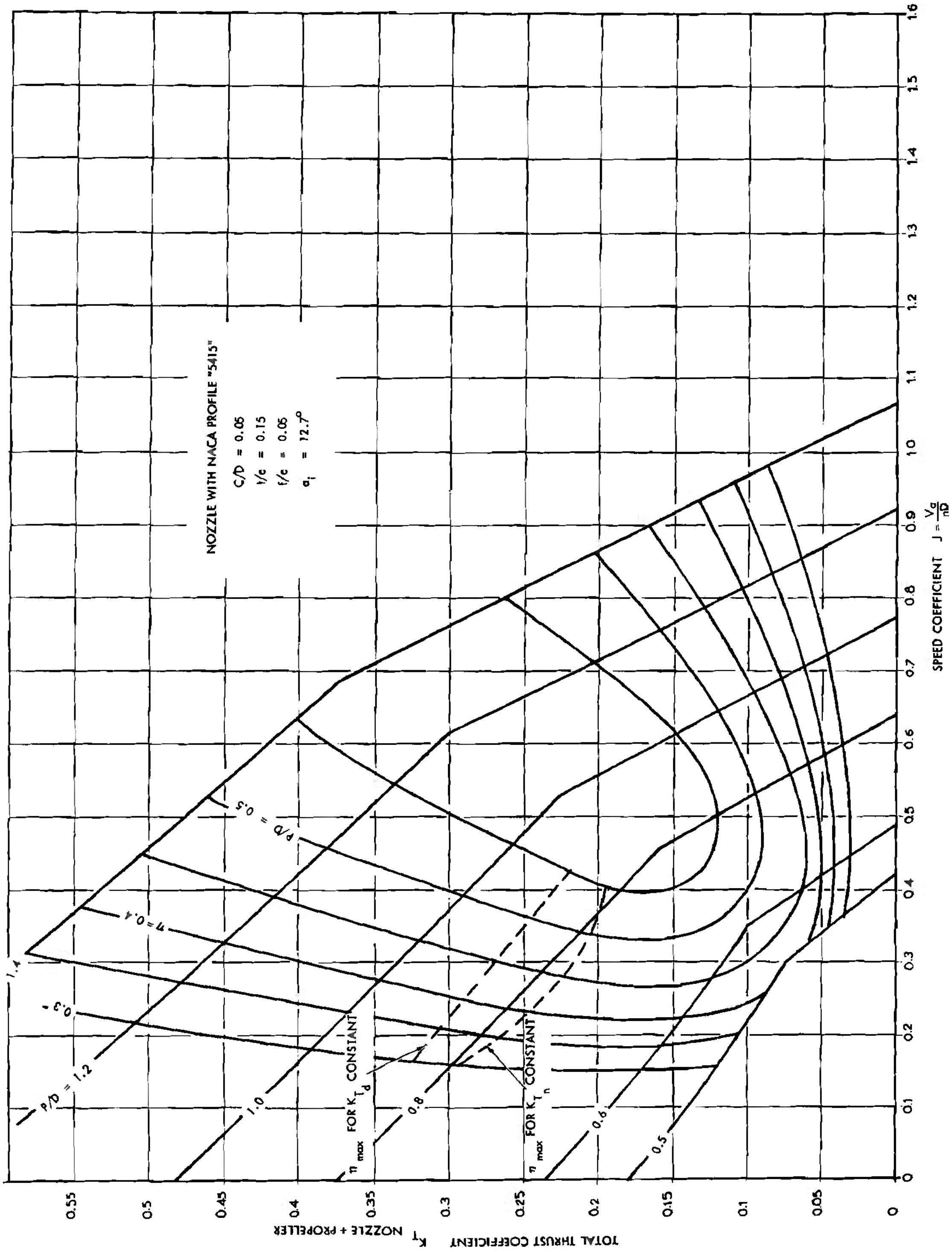


Figure 7-85. Design Chart for Troost B-4.70 Screw in Nozzle

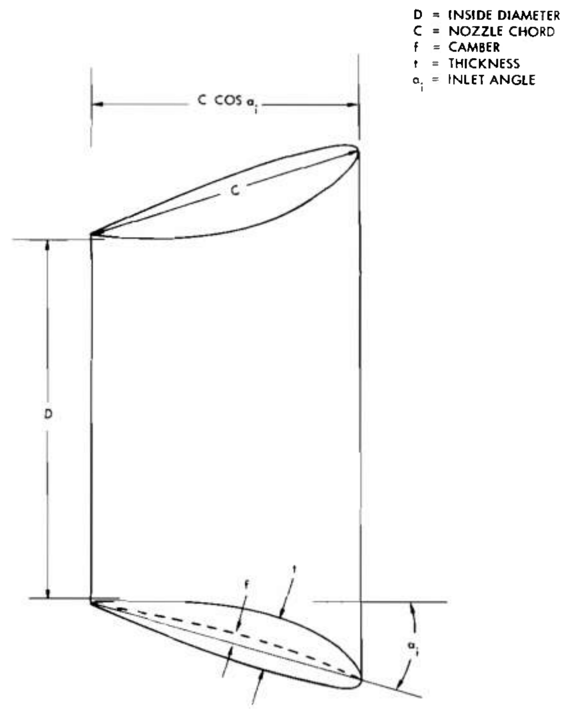


Figure 7-86. Nozzle Definition Sketch

Problem: To find the optimum diameter of the propeller.

Analysis: a. The coefficient K_{T_n} is calculated from the definition

$$K_{T_n} = \frac{K_T}{J^4} = \frac{T_n^2}{\rho V_a^2} = \frac{R\, n^2}{\rho(1-t)(1-w)^4 V_s^4} \quad (7\text{-}97)$$

where units are consistent so that K_{T_n} is dimensionless.

A fourth order parabola is drawn from $K_T = J^4 K_{T_n}$ and the intersection with the "η_{max} for K_T constant" curve determines the design point.

b. The diameter is calculated from the value of J read off:

$$D = \frac{V_s(1-w)}{Jn}\text{, ft} \quad (7\text{-}98)$$

Delivered power is calculated by Eq. 7-96.

Problem No. 3

Given: The delivered power DHP at design rpm are specified for a particular engine. Diameter is fixed and resistance R is given as a function of ship speed. The hull factors t, w, and η_r are also given as functions of speed.

Problem: To find the pitch, efficiency, and highest speed obtainable under the given conditions.

Analysis: a. K_T and J are calculated for several speeds

$$J = \frac{V_s(1-w)}{nD} = \frac{V_a}{nD}\text{, nd} \quad (7\text{-}99)$$

$$K_{T_{(1)}} = \frac{R}{\rho(1-t)n^2D^4} = \frac{T}{\rho n^2 D^4}\text{, nd} \quad (7\text{-}100)$$

b. A point is plotted for each set of K_T and J on the appropriate design chart, and corresponding values of efficiency η_o are found by interpolation. These values are used to calculate K_T for each speed from the relation

$$K_{T_{(2)}} = \frac{DHP\ 550\ \eta_r \eta_o}{V_s(1-w)\rho n^2 D^4}\text{, nd} \quad (7\text{-}101)$$

c. Values of K_T are also plotted with corresponding values of J and curves are drawn through points of $K_{T_{(1)}}$ and $K_{T_{(2)}}$ (see Figs. 7-74 through 7-85). The intersection of these two curves determines the design point. The pitch ratio and efficiency at the design point are found by interpolation. The ship speed is calculated from the speed coefficient J at the design point.

7-7.5.3 Cavitation Diagram

The cavitation diagram, Fig. 7-87, is taken from Ref. 86. The data are for screws designed according to vortex theory. However a 25 percent safety margin has been applied to the data so that the chart is conservative and may be used for an approximate determination of the area ratio EAR (A_E/A in the diagram) for screws of the Troost B-Series. The cavitation number in this diagram is the section cavitation number (plus 25% allowance) at the 0.8 radius at minimum immersion using the dynamic pressure q at the section, thus

$$\sigma_{0.8} = \frac{p_{0.8} - p_v}{q_{0.8}} = \frac{p_{0.8} - p_v}{\frac{1}{2}(\rho V_a^2)\left[1 + \left(\frac{2.51}{J}\right)^2\right]}\text{, nd} \quad (7\text{-}102)$$

where

$$q_{0.8} = ½\left(\rho V_a^2\right)\left[1 + \left(\frac{2.51}{J}\right)^2\right]$$

The ordinate is the thrust normalized by the projected blade area A_p and the resultant velocity at the 0.8 radius:

$$\frac{T}{A_p q_{0.8}} = \frac{T}{½(\rho V_a^2)\left[1 + \left(\frac{2.51}{J}\right)^2\right]A_p}, \text{nd} \qquad (7\text{-}103)$$

The projected area A_p is related to the expanded area A_E by the approximate relation

$$A_p \approx A_E(1.067 - 0.229\,P/D), \text{ft}^2 \qquad (7\text{-}104)$$

Eqs. 7-102 and 7-103 are used to plot a point on Fig. 7-87. If the point falls above the upper limit (Burrill line for heavily-loaded propellers), the area A_p must be increased by an increase in diameter and/or expanded area ratio EAR. If the point falls below the Burrill line and on the right side of the appropriate data, the propeller will be free of cavitation.

This cavitation diagram should not be considered as a design chart because it does not provide a unique solution for any given problem. The chart should be used as a check on cavitation in the design procedure. The Burrill line may be used to estimate the minimum projected area from values of thrust T, advance speed V_a, and estimates of the speed coefficient J.

The thrust deduction and wake factor of the nozzle must first be calculated in order to use the cavitation diagram for shrouded propellers. Figs. 7-88 to 7-91 given the thrust produced by the nozzle for the propeller-nozzle combinations in Figs. 7-82 to 7-85. The speed coefficient J, pitch ratio P/D, and thrust coefficient K_T for the nozzle + propeller are taken from the design point and the thrust coefficient produced by the nozzle is read at corresponding J and P/D. The nozzle thrust deduction θ_n is then calculated:

$$1 - \theta_n = \frac{K_{T_{\text{nozzle + propeller}}}}{K_{T_{\text{propeller}}}} \qquad (7\text{-}105)$$

where

$$K_{T_{\text{propeller}}} = K_{T_{\text{nozzle + propeller}}} - K_{T_{\text{nozzle}}}$$

The value of $K_{T_{propeller}}$ is then entered (with the pitch ratio) into the appropriate open water characteristic curve (Figs. 7-73 to 7-81) to find $J_{open\ water}$. With this value, the nozzle wake fraction Ψ_n is calculated:

$$I - \Psi_n = \frac{J_{open\ water}}{J} \qquad (7\text{-}106)$$

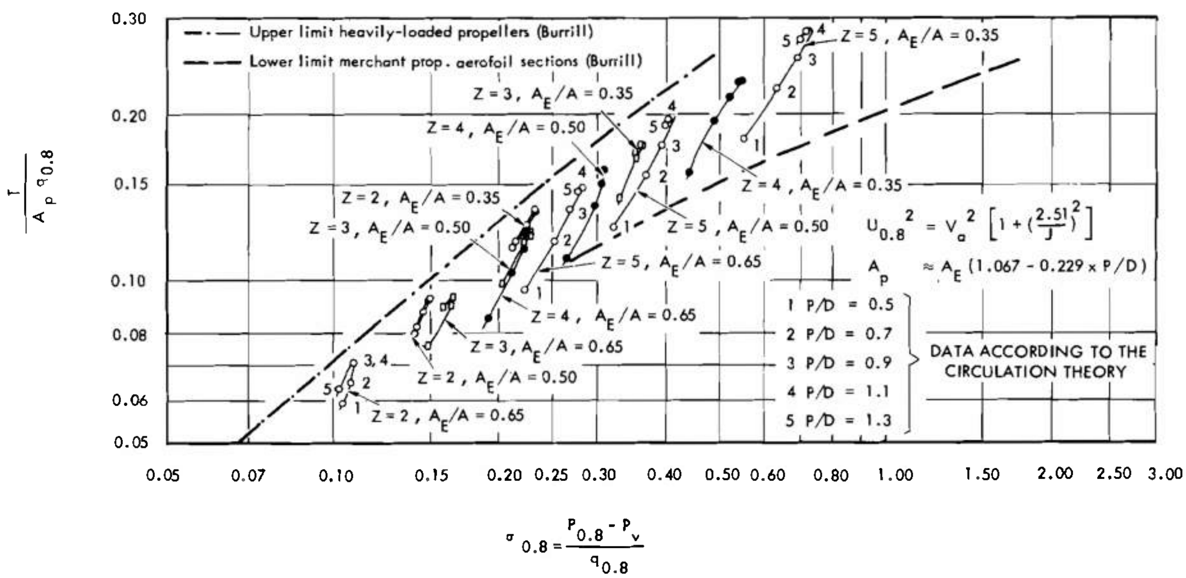


Figure 7-87. Cavitation Diagram for Screw Designed According to the Circulation Theory for a Homogeneous Velocity Field

Now we can enter the cavitation diagram, Fig. 7-87, with the values of

$$\frac{T}{q_{0.8}A_p(1-\theta_n)} \text{ and } \frac{P_{0.8}-p_v}{q_{0.8}} \qquad (7\text{-}107)$$

where

$$q_{0.8} = \tfrac{1}{2}(\rho V_a^2)\left[(1-\Psi_n)^2 + \left(\frac{2.51}{J}\right)^2\right] \qquad (7\text{-}108)$$

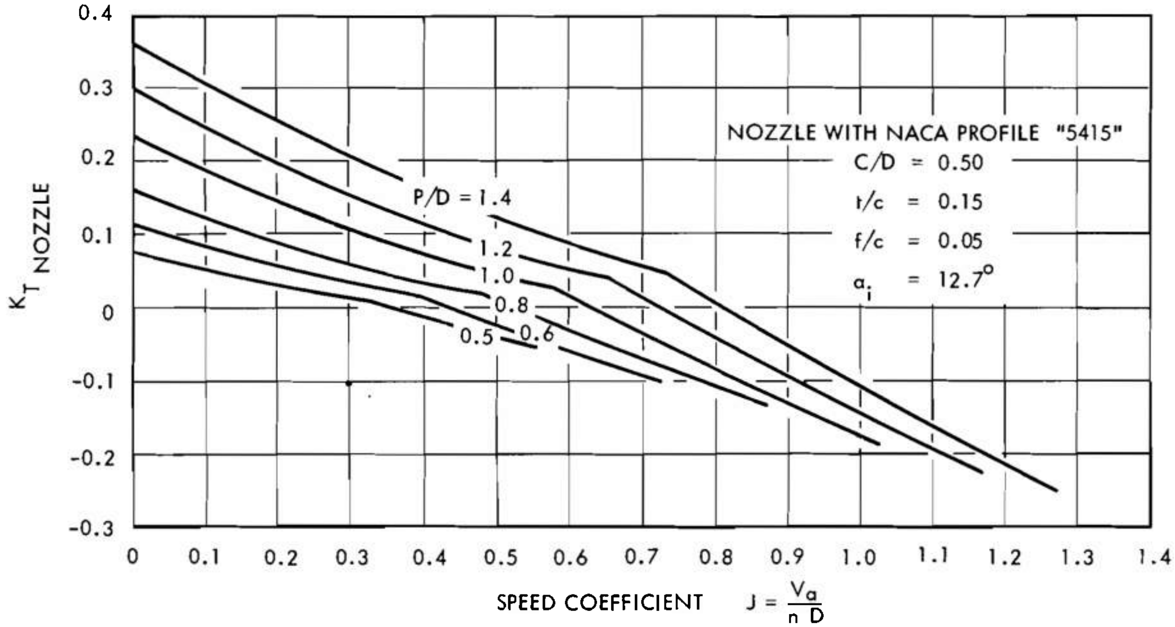


Figure 7-88. Axial Force on Nozzle With the B-3.50 Screw Series

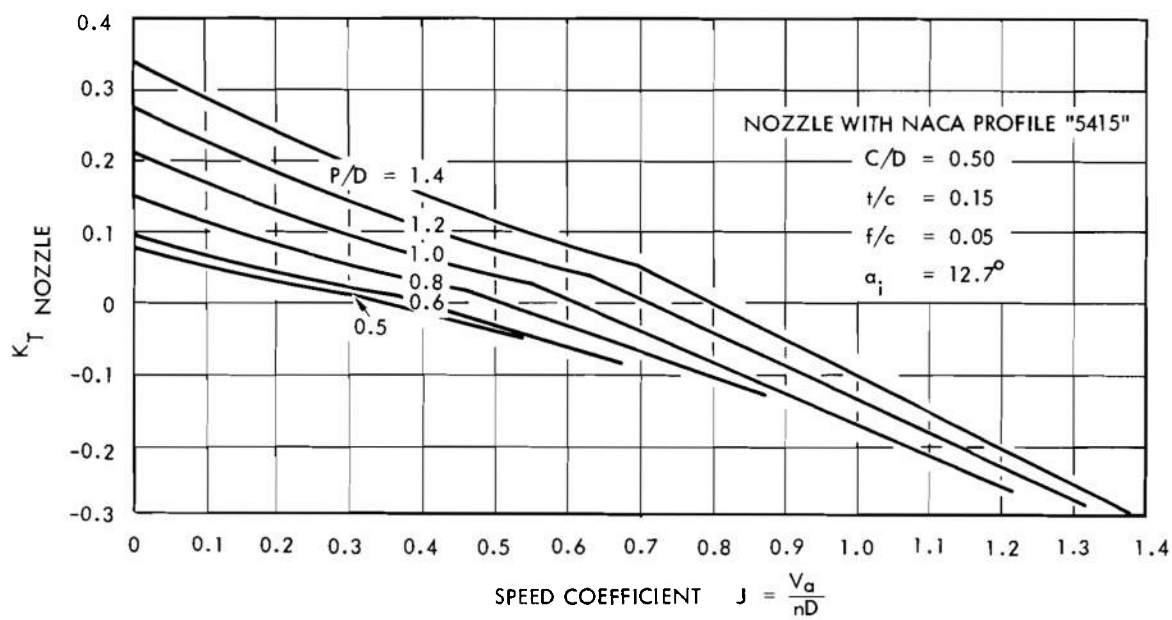


Figure 7-89. Axial Force on Nozzle With the B-4.40 Screw Series

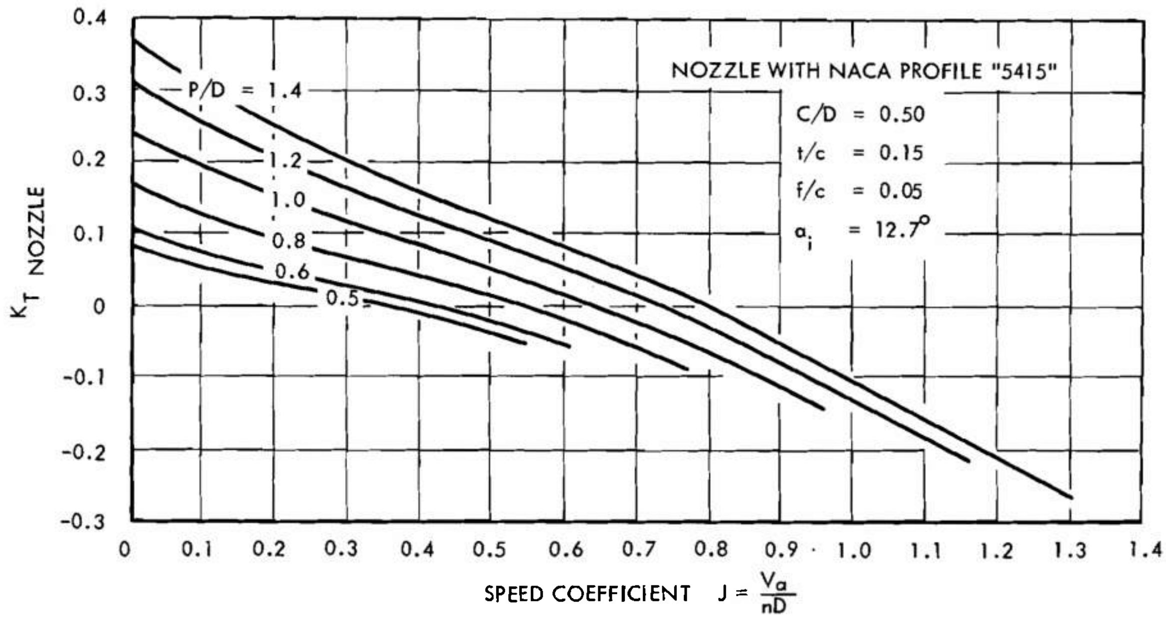


Figure 7-90. Axial Force on Nozzle With the B-4.55 Screw Series

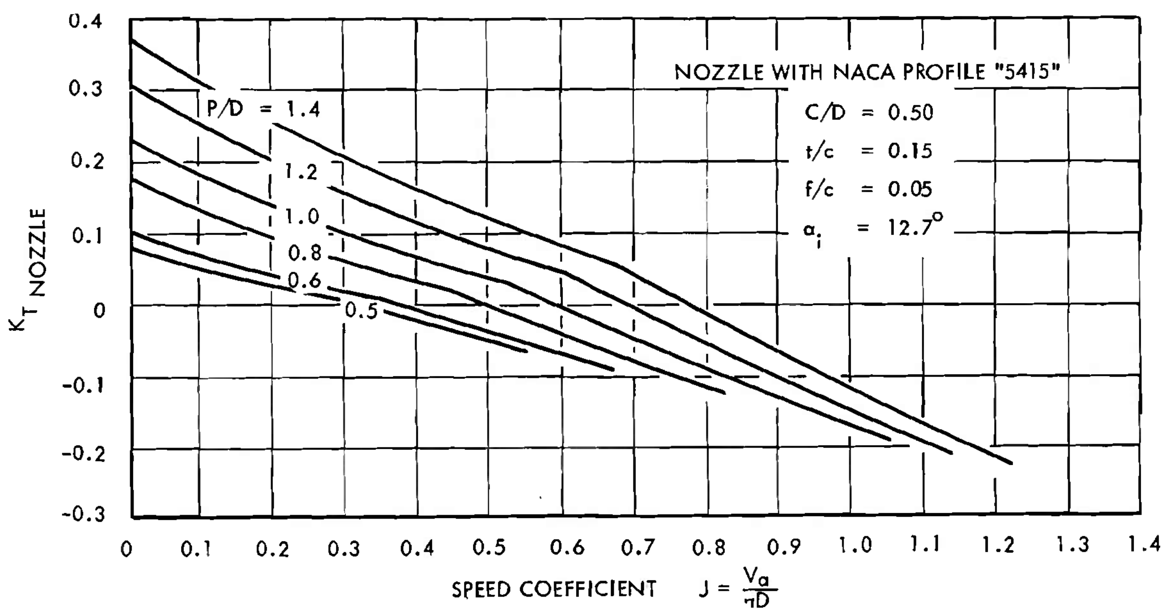


Figure 7-91. Axial Force on Nozzle With the B-4.70 Screw Series

7-7.5.4 Strength

The propeller must have sufficient strength to stand up under normal service conditions. It is important that the minimum blade thickness for adequate strength be chosen from a standpoint of efficiency and weight. It is common practice to calculate the thickness at a typical section near the root and use a linear thickness distribution between this section and the blade tip to determine the thickness at any section (the thickness of any section is defined as the thickness at the point of maximum thickness of that section profile). Some reduction in weight may be realized by making strength calculations at a number of sections to determine the minimum allowable thickness at each section.

The formula which follows is found in the American Bureau of Shipping (ABS) Rules. The derivation of this formula is given in Ref. 88. It is used to calculate the minimum acceptable thickness at the 25 percent radius. The nomenclature used in the ABS Rules is not consistent with this handbook. However, ABS symbols have been retained for this formula.

$$t = 41\left(\frac{AH}{BRN}\right)^{1/2} \pm \frac{1.72CK}{B} \qquad (7\text{-}109)$$

where

t = thickness of blade at one-quarter radius, in.

H = SHP at maximum continuous rating, hp

R = rpm at maximum continuous rating

N = number of blades

$P_{0.25}$ = pitch at one-quarter radius divided by propeller diameter, nd

$P_{0.7}$ = pitch at seven-tenths radius divided by propeller diameter, nd

W = width of blade at one-quarter radius, in.

a = expanded blade area divided by disc area, nd

D = propeller diameter, ft

K = rake of propeller blade, in./ft, multiplied by $D/2$ (with forward rake, use minus sign in formula; with aft rake, use plus sign), in.

$$A = 1.0 + \frac{6.0}{P_{0.7}} + 4.3P_{0.25}$$

$$B = (1 + 1.5P_{0.25})(Wf - C)$$

$$C = \frac{4300Wa}{N}\left(\frac{R}{100}\right)^2\left(\frac{D}{20}\right)^3$$

f,w = material constants from Table 7-43.

TABLE 7-43. REPRESENTATIVE PROPELLER MATERIALS

		f	w
Type 1	Manganese bronze	68	0.30
Type 2	Nickel-manganese bronze	73	0.29
Type 3	Nickel-aluminum bronze	85	0.27
Type 4	Manganese-nickel-aluminum bronze	85	0.27
Type 5	Cast iron	25	0.26

For an example, consider a cast iron Troost B-4.55 series propeller with 3.0 ft diameter and 3.0 ft pitch turning 400 rpm at an advance speed of 13.7 fps (9.3 mph). The calculations are made as follows:

$$J = \frac{V_a}{nD} = \frac{13.7}{3.0\left(\frac{400}{60}\right)} = 0.687$$

$$P/D = 3.0/3.0 = 1.0$$

$$\left.\begin{aligned} K_T &= 0.19 \\ \eta_o &= 0.65 \end{aligned}\right\} \text{Read off Fig. 7-78}$$

$$K_Q = \frac{K_t J}{\eta_o 2\pi} = \frac{(0.19)(0.687)}{(0.65)2\pi} = 0.032$$

$$\rho = 1.99,\ \text{slug/ft}^3\ \text{(salt water)}$$

$$H = \frac{2\pi Qn}{550} = \frac{2\pi K_Q n^3 D^5 \rho}{550}$$

$$= \frac{2\pi(0.032)\left(\frac{400}{60}\right)^3 3^5(1.99)}{550} = 52.5\ \text{hp}$$

R = 400 rpm

N = 4

$P_{0.25}$ = 0.855 from Fig. 7-72

$P_{0.7}$ = 1.0

a = 0.55

$$W = 12 \text{ (factor for 0.25 radius) } a \text{ (factor for 4-blades) } D$$
$$= 12\ (0.793)\ 0.55\ (0.547)\ 3.0 = 8.6, \text{ in. (from Table 7-42)}$$

$$K = 12\ (\tan 15°)\ \frac{3.0}{2} = 4.83$$

$$A = 1.0 + \frac{6.0}{1.0} + 4.3(0.855) = 10.68$$

$$C = \frac{4300}{400}\ (0.26)0.55\left(\frac{400}{100}\right)^2\left(\frac{3}{20}\right)^3 = 0.082$$

$$B = [1 + (1.5 \times 0.855)]\ [(8.6)25 - 0.082] = 490$$

$$t = 41\left[\frac{(10.68)52.5}{490(400)4}\right]^{1/4} + \frac{1.72(0.082)4.83}{490}$$
$$= 1.10$$

$$(t/D)_{0.25} = \frac{1.10}{36} = 0.031$$

With linear thickness distribution, the thickness at any section can be calculated from the thickness at the blade tip $t_{1.0}$ and hub center t_o.

$$t_{r/R} = t_{1.0}(r/R) + t_o(1 - r/R) \qquad (7\text{-}110)$$

where

r = section radius
R = propeller radius

Assuming the tip thickness $t_{1.0}$ to be 0.3 percent of the diameter, the blade thickness fraction $BTF = (t/D)_o$ of the propeller is

$$BTF = \frac{t_{0.25} - r/R\ t_{1.0}}{D(1 - r/R)} = \frac{0.031 - 0.25(0.003)}{0.75}$$
$$= 0.040$$

This is slightly below the standard thickness ($0.045D$) for Troost B-Series four-bladed propellers.

Any increase in the blade thickness for Troost B propellers (over the values given in Table 7-41) should be accompanied by decreases in the pitch and efficiency. As a rule of thumb, the pitch ratio P/D and efficiency η should be reduced by a factor of 0.1 and 0.5, respectively, times the increase in thickness $\Delta t/t$. Thus, if thickness were increased by 40 percent, pitch should be reduced by 4 percent and efficiency by 2 percent.

7-7.5.5 Towing or Overload Conditions

After a propeller has been chosen for the design point of the amphibian, it may be desired to estimate the speed at which a disabled sister ship might be towed or the reduction in speed due to fouling of the hull. The engine characteristics and loading must be given, i.e., available horsepower as a function of rpm and the total drag as a function of ship speed. The wake fraction and thrust deduction may be affected by fouling and thrust loading, respectively. Estimates of these effects should be made. With this information, the following procedures can be executed. An auxiliary plot of $K_{T_d} = K_T/J^2$ versus J can be made from the open water characteristics of the propeller in question. It will be convenient to use semi-log paper for this curve. Now, several values of speed are selected over the range of the expected solution and K_{T_d} is calculated at each value from Eq. 7-94. The auxiliary plot is entered to find J and, hence, rotative speed n for each speed. The open water efficiency is found for each speed from the corresponding value of J entered in the characteristic curves. Propulsive efficiencies ($\eta_p = \eta_o \eta_h \eta_r$) are computed and then the required delivered horsepower DHP:

$$DHP = \frac{R_T V_s}{550\eta_p}, \text{ hp} \qquad (7\text{-}111)$$

where

R_T = total resistance, lb
V_s = ship speed, ft/sec

Finally, a plot is made with speed, required DHP, and available horsepower as functions of rotative speed n. The desired speed corresponds to the point n at which the two horsepower curves cross.

Some minimum static thrust or bollard pull produced by the propulsor may also be required. The static thrust (thrust at zero advance speed) of a propulsion system is a function of both the engine and propulsor characteristics. Their torque characteristics, together, determine the maximum rpm which can be attained with zero advance speed and thus the power absorbed in this condition. The bollard pull is a function of

the static thrust, and the interaction of hull and propulsor given by the zero speed thrust deduction. A simple test may be included with model self-propulsion tests to determine both the propeller thrust and the bollard pull. Otherwise, the propeller open water characteristics may be used to estimate the static thrust as follows. The thrust and torque coefficients (K_{T_s} and K_{Q_s}) are obtained from the characteristics curves at $J = 0$. Torque is plotted as a function of rpm from the relation

$$Q_s = (\rho K_{Q_s} D^5) n^2 \text{, ft-lb} \qquad (7\text{-}112)$$

The engine torque curve is also plotted, and the intersection of the two lines gives the desired torque which is used to calculate the static thrust T_s

$$T_s = \frac{K_{T_s} Q_s (1 - t_s)}{K_{Q_s} D} \text{, lb} \qquad (7\text{-}113)$$

where t_s = estimated zero speed thrust deduction.

7-7.5.6 Astern Thrust

Astern thrust is important for maneuvering and stopping. The reach (stopping distance) of a vessel is a function of both the propeller characteristics and the engine-transmission characteristics. Propeller characteristics for a model series operating both ahead and astern are reported in Ref. 89. An analytical method for calculating the reach is presented in Ref. 90. It should be noted that these calculations are rather complex. Consideration might be given to including stopping tests with the self-propulsion studies. At any rate, while a short reach is desirable, the stopping distance of the amphibian is not generally given as a design requirement.

7-7.5.7 Service Allowance

The service shaft horsepower is the power required to maintain an average (design) speed over a long period of time (life of the vessel). It is the power predicted from the model tests augmented by a percentage known as the service allowance. It includes the power necessary to offset the effects of wind and waves, rolling, random steering about the straightline course, fouling of the bottom, and losses in propeller efficiency due to increased loading and roughening of the blades. The service allowance is like a factor of safety. A good value for the service allowance for amphibians is around 25 percent. This value may be modified with increased design experience.

7-7.5.8 Numerical Examples

(1) Given:

Design Speed V_s	=	9 mph (13.2 ft/sec)
Resistance R_T	=	2,900 lb
Diameter D	=	2.5 ft
$1 - t$	=	0.65; thrust deduction $t = 0.35$
$1 - w$	=	0.75; wake fraction $w = 0.25$
η_r	=	0.96
ρ	=	1.99 slug/ft^3

Find: Optimum rpm, pitch, and required horsepower for a B-4.55 propellerr(4-bladed with *EAR* = 0.55)

a. From: Eq. 8-94:

$$K_{T_d} = \frac{2900}{1.99(0.65)(0.75)^2(13.2)^2(2.5)^2} = 3.62$$

$K_T = K_{T_d} J^2$ so a parabola may be constructed on Fig. 7-78.

Points for parabola:

J	K_T
0.2	0.145
0.25	0.226
0.3	0.326
0.35	0.445

At the intersection of the parabola and the curve "η_{max} for K_{T_d} constant" read off the following design information.

Design Point:

K_T = 0.255
J = 0.265
η_o = 0.33
P/D = 0.778 (Pitch = 1.94 ft)

b. Eq. 7-95 yields,

$$n = \frac{13.22(0.75)}{0.265(2.5)} = 15.0 \text{ rps} \quad (900 \text{ rpm})$$

c. Eq. 7-96:

$$DHP = \frac{2900(0.75)13.2}{(0.65)0.96(0.33)550} = 254 \text{ hp}$$

Cavitation check

Assume depth of 1.0 ft at 0.8 radius, then

$$P_{0.8} = P_{atmospheric} + P_{head}$$
$$= 2116 + 64 = 2180 \text{ lb/ft}^2$$
$$p_v = 0.98\,(0.25)\,144$$
$$= 35 \text{ lb/ft}^2 \text{ (salt water at 59°F, Table 7-40)}$$

From Eq. 7-102,

$$q_{0.8} = \tfrac{1}{2}\rho V_a^2 \left[1 + \left(\frac{2.51}{J}\right)^2\right]$$
$$= \frac{1.99}{2}\,[13.2^*(0.75)]^2 \left[1 + \left(\frac{2.51}{0.265}\right)^2\right]$$
$$= 8{,}840 \text{ lb/ft}^2$$

*Note, $V_a = V_s\,(1 - w)$

Eq. 7-102:

$$\sigma_{0.8} = \frac{2180 - 35}{8{,}840} = 0.243$$

$$A_E = \left(\frac{\pi D^2}{4}\right) EAR$$

From Eq. 7-104

$$A_p \approx \frac{\pi (2.5)^2\, 0.55}{4}\,[(1.067 - 0.229)(0.778)]$$

(since EAR = 0.55 as given)

$$\approx 2.7(0.887) \approx 2.39 \text{ ft}^2$$

$$T = \frac{2900}{(1 - t)} = \frac{2900}{0.65}$$

Eq. 7-103:

$$\frac{T}{q_{0.8}A_p} = \frac{2900}{(0.65)8{,}840(2.39)} = 0.213$$

Plotting this point $[\sigma_{0.8}\ T/(q_{0.8}\,A_p)]$, i.e., (0.243, 0.213) on Fig. 7-87 shows that this design lies above this limit for heavily loaded propellers and, therefore, an increase in EAR is in order. $T/q_{0.8}\,A_p$ should be reduced to about 0.14 to insure cavitation-free operation. Thus, the expanded area ratio EAR should be increased to about (0.213/0.14) 0.55 ≈ 0.85 and the design cycle repeated. The changes in rpm, pitch, and power should be small.

(2) Given:

Design Speed V_s	=	7.5 mph (11.0 ft/sec)
Resistance R_T	=	4,200 lb
Diameter D	=	2.5 ft
rpm	=	900 (15 rps)
$1 - t$	=	0.75
$1 - w$	=	0.80
η_r	=	1.0

Find: Optimum diameter, pitch, and required horsepower for a shrouded B-455 propeller (nozzle characteristics the same as in Fig. 7-84) with EAR = 0.55

a. From Eq. 7-97

$$K_{T_n} = \frac{4200(15)^2}{1.99(0.75)(0.80)^4(11.0)^4} = 105.5$$

Since $K_T = K_{T_n}\,J^4$, we can construct a 4th order parabola on Fig. 7-84.

Points for parabola:

J	K_T
0.2	0.169
0.22	0.248
0.24	0.350

At the intersection of this parabola read off the following information at the curve "η_{max} for K_{T_n} constant", design point

$$K_{T_{nozzle + propeller}} = 0.274$$
$$\eta_o = 0.40$$
$$J = 0.226$$
$$P/D = 0.82$$

b.

$$D = \frac{V_a}{nJ} = \frac{V_s(1 - w)}{nJ} = \frac{11.0(0.80)}{15.0(0.226)}$$
$$= 2.6, \text{ ft}$$

(since $V_a = V_s(1 - w)$ by Eq. 7-82)

$$P = (0.82)2.6 = 2.10, \text{ ft}$$

c. Eq. 7-96:

$$DHP = \frac{4200(0.80)11.0}{0.75(1.0)0.40(550)} = 280 \text{ hp}$$

Cavitation check

Since the propeller is in a nozzle, the performance factors of the nozzle must first be determined from Fig. 7-90:

$$K_{T_{nozzle}} = 0.095$$

Eq. 7-105:

$$K_{T_{propeller}} = K_{T_{nozzle\ +\ propeller}} - K_{T_{nozzle}}$$

$$= 0.274 - 0.095 = 0.179$$

Eq. 7-105:

$$1 - \theta_n = \frac{K_{T_{nozzle\ +\ propeller}}}{K_{T_{propeller}}} = \frac{0.274}{0.179} = 1.53$$

Enter Fig. 7-88 with $K_T = 0.179$ and $P/D = 0.82$ to find:

$$J_{open\ water} = 0.50$$

Eq. 7-106:

$$1 - \Psi_n = \frac{0.50}{0.226} = 2.21$$

Eq. 7-108:

$$q_{0.8} = \frac{1.99}{2}\,[11.0(0.80)]^2 \left[(2.21)^2 + \left(\frac{2.51}{0.226}\right)^2\right]$$

$$= 9{,}850 \text{ lb/ft}^2$$

Using the same values for stagnation pressure and vapor pressure as in the previous example,

Eq. 7-102:

$$\sigma_{0.8} = \frac{p_{0.8} - p_v}{q_{0.8}} = \frac{2{,}145}{9{,}850} = 0.218$$

Eq. 7-104:

$$A_p \approx A_E(1.067 - 0.229\,P/D)$$

$$\approx \pi\,\frac{(2.5)^2}{4}\,0.55\,[1.067 - 0.229(0.82)]$$

$$\approx 2.56 \text{ ft}^2$$

Eq. 7-108:

$$\frac{T}{q_{0.8}(1-\theta_n)A_p} = \frac{4{,}200}{(0.75)9{,}850(1.53)2.56}$$

$$= 0.145$$

(since $T = R/(1-t)$ by Eq. 7-83)

Once again, the point $[\sigma_{0.8}, T/[q_{0.8}(1-\theta_n)A_p]]$, i.e., (0.218, 0.145) lies slightly above the upper limit in Fig. 7-87 indicating that a slight increase in blade area is in order if this type of propeller is used.

(3) Given: B-4.55 screw ($Z = 4$, $EAR = 0.55$)

Diameter = 2.5 ft

DHP = 130 at 700 rpm and the following data (from LARC V, Table 7-35)

Ship Speed, mph	7	8	9
ft/sec	10.3	11.7	13.2
Resistance, lb	1600	2340	2910
$1 - t$	1.09	1.09	1.08
$1 - w$	1.14	1.175	1.168
η_r	0.96	0.93	0.93

Find: Optimum pitch and highest possible speed. From Eqs. 7-99, 7-100, and 7-101 find corresponding values of:

J	0.402	0.474	0.530
$K_{T_{(1)}}$	0.139	0.203	0.254
η_o	0.515	0.530	0.535
$K_{T_{(2)}}$	0.285	0.241	0.218

Now, plot values of $K_{T_{(1)}}$ and $K_{T_{(2)}}$ on Fig. 7-78 versus J.

The intersection gives the design point at $J = 0.50$ and $P/D = 0.92$.

$$\text{Maximum speed } V_s = \frac{JnD}{(1-w)} = 0.50\left(\frac{700}{60}\right)\frac{2.5}{1.17}$$

$$= 12.5 \text{ ft/sec} = 8.50 \text{ mph}$$

where $(1 - w)$ is found by interpolation

(4) Given: The same engine and propeller as in example (3) above. Assume engine torque is constant with respect to rpm.

$1 - t_s = 0.90$ (at zero ship speed)

Find: Static thrust and bollard pull.

with P/D = 0.92,

$K_{T_{J=0}}$ = 0.395 (from Fig. 7-78)

$K_{Q_{J=0}}$ = 0.50 (from Fig. 42 of Ref. 86)

$$Q = \frac{DHP\,550}{2\pi n} = \frac{130(60)550}{2\pi 700}$$

$$= 975 \text{ ft-lb}$$

Eq. 7-113:

$$T_s = \frac{(0.395)975(0.90)}{(0.050)2.5}$$

$$= 2770 \text{ lb (bollard pull)}$$

Eq. 7-112:

$$n = \left(\frac{975}{1.99(0.050)(2.5)^5}\right)^{1/2} = 10.0 \text{ rps (600 rpm)}$$

Note that these estimates do not take into account the effect that the hull may have on the propeller inflow and hence, on rotative speed, and static thrust.

7-7.5.9 Geometry of the Screw Propeller (Drawings)

Standing behind a ship and looking forward, one sees the face (high-pressure surface) of the propeller blade. The other side is the back (low-pressure surface). These are both complex warped surfaces. In the simplest case the face lies on a helicoidal surface which is generated by a radial line rotating at constant velocity about an axis and simultaneously moving along this axis at a constant velocity (Fig. 7-92). The axial distance covered in one revolution is the pitch P. If the pitch is allowed to vary with radius, the surface is no longer a true helicoid. In Fig. 7-93 the pitch at the hub is less than at the tip and the pitch distribution is linear. The radial pitch distribution of a propeller, however, is not restricted to being constant nor linear.

Many propellers are designed so that the nose-tail line of each section lies on the reference helicoidal surface. In this case the pitch at each section is the true pitch of the propeller. In the case of the Troost series propellers, the face lies on the reference surface (the sections are "flat-faced") and the pitch of the face at each section is called the nominal pitch.

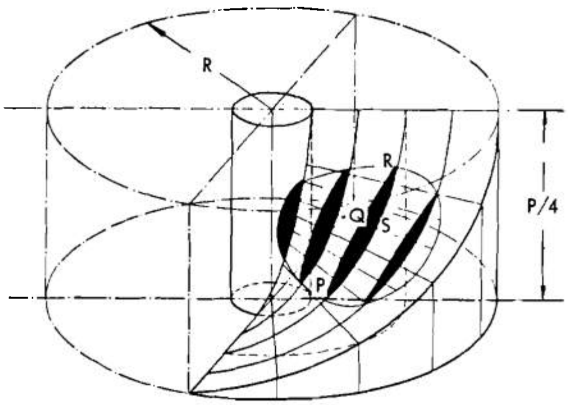


Figure 7-92. Helicoidal Surface With Constant Pitch

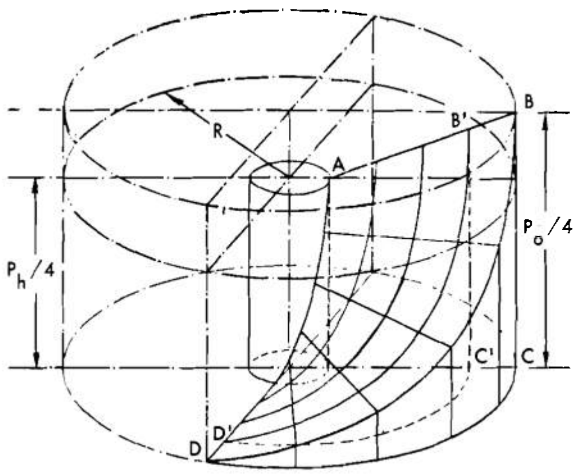


Figure 7-93. Helicoidal Surface With Radially Varying Pitch

Drawings of the propellers are restricted to plane surfaces, so certain conventions are used in delineating a propeller on paper. Two important reference lines are the screw axis and the generatrix (also called the blade reference line). The three projections of the blade show the actual outline of the propeller blade as viewed from above, the side, and from behind. The first projection is often omitted from propeller drawings. The projected area of the blade is the area enclosed by the projected outline in the second projection (see Fig. 7-94) exclusive of the hub.

The developed and expanded outlines are conventions which are imaginary in that they

cannot be seen on an actual propeller. The developed outline is obtained by rotating each section about the generatrix so that the nose-tail line (or the nominal face pitch line) lies in the plane of the paper in the second projection. The angle of rotation for each section is the pitch angle θ_r of that section:

$$\theta_r = \mathrm{Tan}^{-1}\left(\frac{P_r}{2\pi r}\right) \qquad (7\text{-}114)$$

where

r = radius of the section, ft

P_r = pitch at that particular section, ft

The expanded outline is of greater importance and is generated by "unrolling" the developed sections into straight lines. The section shapes are drawn in the expanded outline at several radii. The expanded area A_E is simply the area enclosed by the expanded outlines of all the blades.

If the pitch is not constant, the radial variation is shown as in Fig. 7-72. A blade skew line and maximum thickness line are also seen in expanded outlines in this figure. The projected outline is omitted from the third projection in this figure which merely shows the radial distribution of maximum thickness and the extent of blade rake.

7-7.6 MODEL TESTS

Model propeller testing is an important phase in the design of any ship. Open water tests are used to confirm the propulsion characteristics predicted by a detailed design study. Design charts are based on the test results for a series of models. Self-propulsion tests are required to find the hull efficiency η_h of the propeller-hull combination in cases where past experience does not provide sufficient data to make a good estimate. Cavitation testing may be indicated for designs requiring quiet operation, freedom from cavitation damage, or where heavy loadings may result in thrust breakdown.

There are many factors which favor model testing over full scale or prototype testing in the early stages of a design. Eventually, of course, a prototype vehicle must be tested to show that the performance requirements of the design have been fulfilled. Savings in both time and money are the main advantages afforded by scale model tests. In addition, unwanted effects from wind, waves, tides, etc., are avoided by testing under the controlled conditions of a towing tank. Dynamometry and data are more reliable and accurate in a controlled environment.

The scaling laws which apply to propeller model testing will be discussed before describing the various tests and their usage.

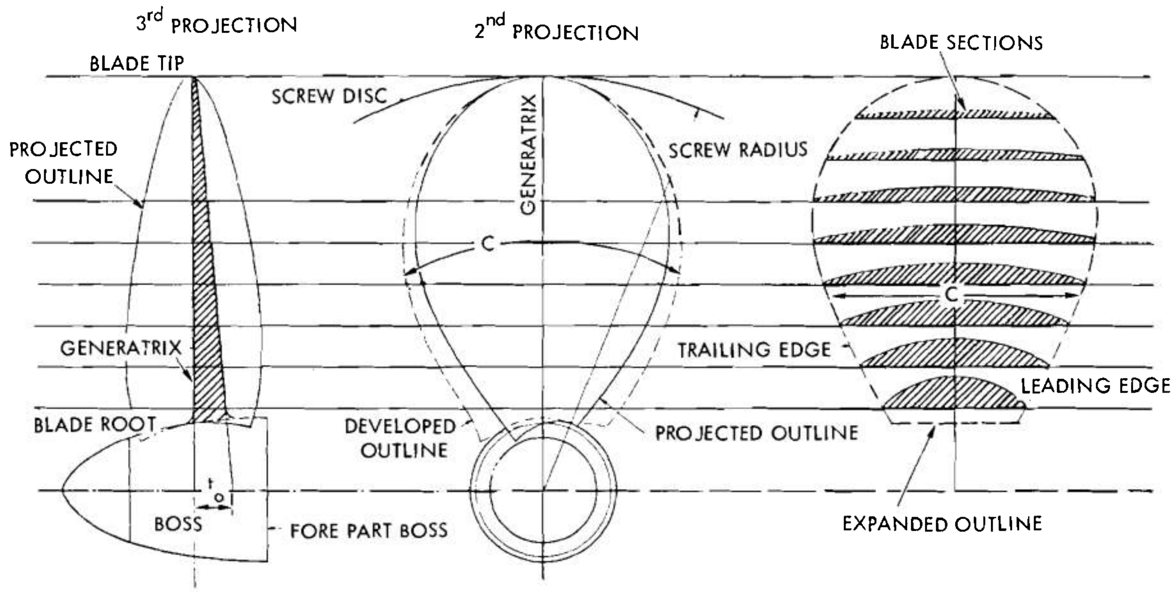


Figure 7-94. Various Projections of a Screw

7-7.6.1 Laws of Similitude

Propeller model tests are governed by laws of similitude or scaling laws in a similar manner to ship model tests. First, the model must be geometrically similar to the prototype propeller with all dimensions reduced by the scale ratio λ; thus the diameter is

$$D_m = D_p/\lambda \qquad (7\text{-}115)$$

where the subscripts m and p refer to model and prototype, respectively. The blade thickness, chord, pitch, etc., are all defined in the same manner. Dynamic similitude must be satisfied by operating the model at the same speed coefficient as the prorotype

$$\left(\frac{V_a}{nD}\right)_m = \left(\frac{V_a}{nD}\right)_p \qquad (7\text{-}116)$$

In order to completely satisfy the conditions for dynamic similitude, the ratio of inertia forces to friction forces and gravitation forces should be made constant by Reynolds scaling and Froude scaling. When the propeller is working at or near the water surface, gravitational forces become important so that the Froude number of the model must be the same as for the prototype, i.e.,

$$\text{Froude number } F_N = \left(\frac{V_a}{\sqrt{gD}}\right)_m = \left(\frac{V_a}{\sqrt{gD}}\right)_p \text{, nd} \qquad (7\text{-}117)$$

where g = gravitational acceleration, ft/sec^2

Combining Eqs. 7-115 and 7-117 (with g constant) we obtain

$$V_{a_m} = V_{a_p}/\sqrt{\lambda} \qquad (7\text{-}118)$$

Similarly, by equating the speed coefficients of model and prototype (Eq. 7-117) we deduce, with the aid of Eqs. 7-115 through 7-118, that

$$n_m = n_p\sqrt{\lambda} \qquad (7\text{-}119)$$

If, for example, one were testing a quarter scale semi-submerged propeller (λ = 4.0), he should run the model at twice prototype rpm (Eq. 7-119) and one-half the speed (Eq. 7-118) of advance of the prototype. In addition, the depth of submergence should be geometrically scaled. Thus, the restriction imposed by Froude scaling is not too serious so that model tests of surface propellers are practical. We shall see, however, that the conditions imposed by Reynolds scaling are not so easily satisfied. Generally, propellers operate deeply submerged so that gravitational forces are insignificant and Froude scaling (Eqs. 7-117 and 7-118) need not be applied in propeller tests.

The Reynolds number Re for a propeller may be written

$$Re = \frac{DV_a}{\nu} \qquad (7\text{-}120)$$

where ν is the kinematic viscosity, ft^2/sec.

To assure that the ratio of friction forces to inertia forces are the same for model and prototype, the Reynolds numbers must be equated. If the viscosity is held constant we can derive:

$$V_{a_m} = V_{a_p}\lambda \qquad (7\text{-}121)$$

and

$$n_m = n_p\lambda^2 \qquad (7\text{-}122)$$

These requirements are difficult if not impossible to fulfill in practice. If we take the same quarter scale model and apply Reynolds scaling, the propeller should be run at sixteen times prototype rpm and four times the advanced speed. It is customary in propeller model testing work to be satisfied with as near an approach to the theoretical requirement as is practical. To this end, the models are made large and tested at the highest speeds obtainable with the available equipment. Fortunately, blade friction has no appreciable influence on the propeller forces above a certain critical value (Ref. 67). Even when the test Reynolds number is below the critical value, the resulting "scale-effect" errors are small and conservative; predicted efficiencies from model tests will be on the low side.

The conclusion that blade friction is usually negligible simplifies matters considerably. In this case only the inertia forces are considered and the necessary condition for similitude, Eq. 7-116, becomes sufficient for dynamic similitude. If gravitational forces from free surface effects are involved; Froude scale, Eq. 7-117, must also be applied.

The laws of similitude must be extended in cases where the propeller is affected by cavitation. The cavitation number σ_o is modeled to assure similarity of the cavitation process (Eq. 7-93). Strictly speaking, Froude scaling should also be applied in this case because local cavitation number on a blade element varies with change in depth due to rotation. In practice, however, this refinement is overlooked with no sensible error being introduced.

7-7.6.2 Open Water Tests

Model propeller open water tests are generally conducted in towing tanks but may also be made in recirculating flow channels in which case the propeller rotates in a stationary plane and the water speed past the propeller determines the advance speed V_a. In a towing tank the propeller is supported by a horizontal drive shaft protruding from a streamlined housing attached to the towing carriage. The housing contains the dynamometer and the drive shaft is made sufficiently long so that the presence of the housing will not affect the inflow to the propeller which advances into undisturbed water. A streamlined fairwater is placed in front of the propeller hub.

Test runs are made over a range of speed coefficient J which is controlled by varying the towing carriage speed (advance speed) and/or the propeller rotative speed rpm. Thrust, torque, advance speed, and rpm are measured for each run. These data are then reduced to coefficient form and presented graphically as open water characteristic curves. The nondimensional coefficients which are most generally used are

$$\left.\begin{aligned} K_T &= \frac{T}{\rho n^2 D^4} \\ K_Q &= \frac{Q}{\rho n^2 D^5} \\ \eta_o &= \frac{TV_o}{2\pi Q n} = \frac{K_T J_o}{K_q 2\pi} \\ J_o &= \frac{V_o}{nd} \end{aligned}\right\} \qquad (7\text{-}123)$$

where the subscript o is used to denote "open water".

The open water characteristics of a typical propeller are presented in Fig. 7-95. The curve for torque coefficient K_Q may be omitted in some presentations because it can be derived directly from the curves of thrust coefficient K_T and efficiency η_o.

7-7.6.3 Self-propulsion Tests

Self-propulsion tests are conducted to determine the interactions between the propeller and hull and in particular the wake fraction, thrust deduction, and relative rotative efficiency. An open water test of the propeller alone and a resistance test of the hull alone are required along with the self-propulsion test in order to obtain the desired results.

The model propeller is located in its proper position on the hull model with all the appendages. The model scale for the propeller and hull must, of course, be the same. An electric motor drives the propeller, and arrangements are made so that thrust, torque, and rpm can be measured. The hull is attached to the towing dynamometer in the same manner as for resistance tests.

The test is conducted as follows: (1) the towing carriage is brought up to the desired speed (Froude scaling), and (2) then the propeller rpm is adjusted until the drag force on the dynamometer is equal to a predetermined value D_f. The thrust, torque, and rpm are recorded. The dynamometer force D_f arises from the difference in friction resistance coefficient between model and prototype (Reynolds scaling effect—see par. 7-4.1.1.3). In practice, the effects of frictional resistance scaling for amphibian models have been neglected because the friction resistance is a small percentage (about 5 percent) of the total resistance. Hence, the dynamometer force for an amphibian self-propulsion test would be set at zero. For the determination of D_f in tests where Reynolds effects are important see Ref. 67, p. 146.

The thrust deduction t is found from a comparison of the measured thrust T necessary to propel the model at a given speed V_s and the resistance R of the hull alone at the same speed

$$(1 - t) = \frac{R}{T} \qquad (7\text{-}124)$$

The efficiency of a propeller η_B behind a model hull generally differs from the efficiency

NUMBER OF BLADES	3
EXP. AREA RATIO	0.559
MWR	0.393
BTF	0.039
P/D	0.804
DIAMETER	7.460 IN.
PITCH	6.000 IN.
ROTATION	R. H.
MAXIMUM HUB DIAMETER	2.51 IN.
MINIMUM HUB DIAMETER	1.66 IN.
HUB LENGTH	2.375 IN.

DESIGNED BY MICHIGAN WHEEL

An 8 - inch M - P type with hub built up by DTMB and diameter cut to 7.46 inches. The final dimensions are given above.

TESTED FOR LYCOMING DIVISION OF AVCO CORPORATION

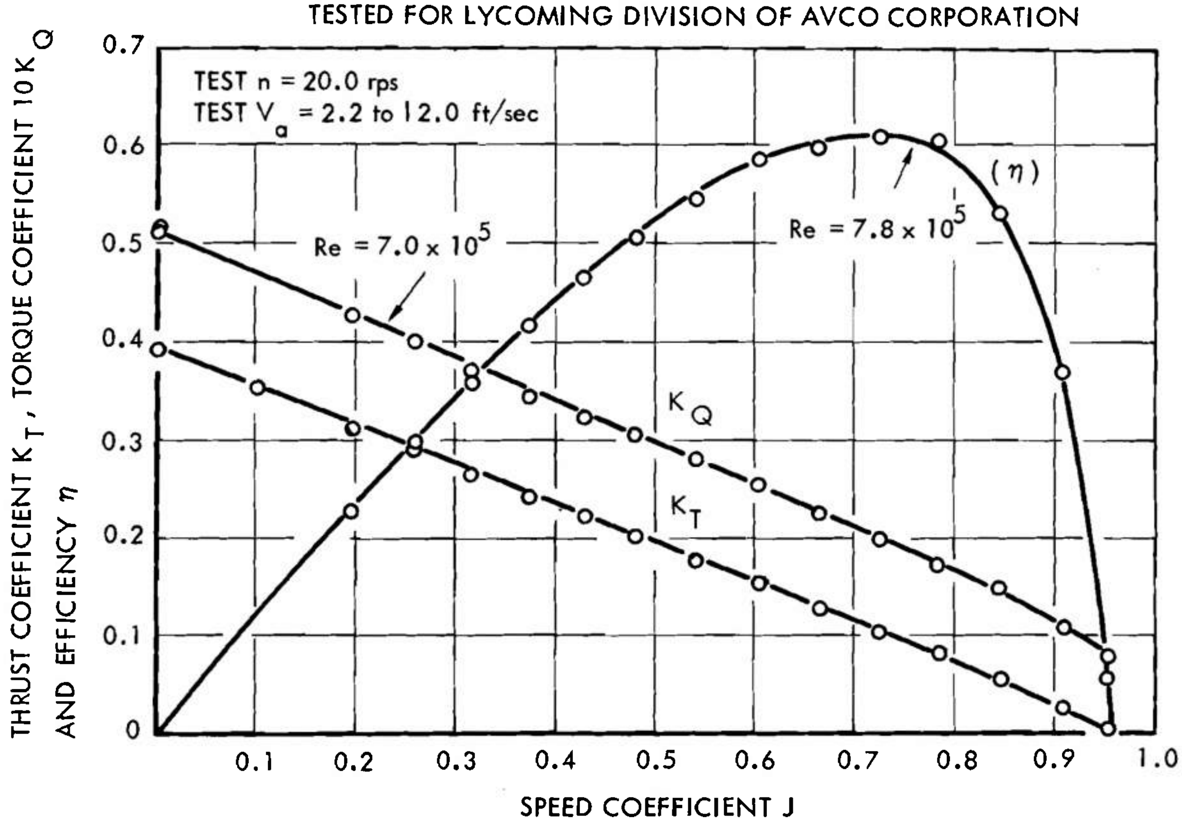


Figure 7-95. Open-water Characteristics for DTMB Propeller 4008

η_o, in open water. They are related by the relative rotative efficiency η_r as in Eq. 7-85. In this equation, the values of the wake fraction w and relative rotative efficiency η_r are not unique. Their values depend upon whether they are determined by thrust or torque identity. With thrust identity the thrust coefficient K_T is calculated from the measured self-propulsion thrust and rpm. With this value of K_T, the open-water characteristic curves are entered to find the speed coefficient J and corresponding open water efficiency η_o, and torque coefficient K_{Q_o}. The advance speed is then calculated

$$V_a = JnD \tag{7-125}$$

The wake is then determined

$$(1 - w) = \frac{V_a}{V_s} \tag{7-126}$$

Finally, the relative rotative efficiency is given by

$$\eta_r = \frac{K_{Q_o}}{K_Q} = \frac{Q_o}{Q} \tag{7-127}$$

With torque identity, a similar procedure is followed, but the open water curves are entered with a calculated (test) value of torque coefficient K_Q which will result in a slightly different value of speed coefficient J and, consequently, η_o. With torque identity, the relative rotative efficiency is redefined by

$$\eta_r = \frac{K_T}{K_{T_o}} = \frac{T}{T_o} \tag{7-128}$$

Thrust identity is more commonly used in this country than torque identity. An alternative method is to use the average speed coefficient found from both thrust and torque identity.

The wake fraction found from self-propulsion tests is known as the effective wake fraction and must not be confused with the "nominal" wake which is obtained from measurements of the velocities in the propeller planes of a towed model without a propeller.

7-7.6.4 Cavitation Tests

Modeling of the cavitation number σ_o necessitates that cavitation tests of a model propeller be conducted in a variable pressure flow channel or water tunnel. In this type of facility a vacuum pump is used to control the ambient air pressure over the free surface. In this manner, the static pressure P_o at the propeller axis can be varied to provide a wide range of cavitation numbers σ_o (see Eq. 7-93) for testing. The facility discussed in Ref. 91 allows model propeller tests to be made for a range of σ_o from about 0.2 to 5.0.

Tests are usually conducted at fixed values of σ_o, i.e., P_o and water speed are held constant. The speed coefficient J is then varied with propeller rpm. The results of propulsion tests can be presented as open water characteristic curves with cavitation number as a parameter as in Fig. 7-71. In addition to measurements of thrust and torque, cavitation inception studies may be made by observing the blade cavities with stroboscopic illumination.

SECTION III SUSPENSION SYSTEM DESIGN

7-8 SUSPENSION SYSTEMS APPROPRIATE FOR USE ON WHEELED AMPHIBIANS

The suspension system provides flexible support between the ground and the amphibian's hull or frame. This support must be flexible in order to reduce the transmission of shock and vibrations to the hull and personnel, arising from encounters with uneven terrain. The maximum overland speed of an amphibian is often limited by the operator's ability or willingness to tolerate large motions and accelerations.

The suspension system provides the necessary clearance between the ground and hull in order to enable the amphibian to traverse uneven terrain and to surmount obstacles. It also has an important effect on the stability of the amphibian on side slopes.

The suspension system provides the contact area between the amphibian and the ground. Mobility in soft soils is a function of the ground pressure exerted by the tires. Hence, both the gross vehicle weight and the total ground contact area provided by the suspension system help to determine the amphibian's mobility.

The principal components of suspension systems for wheeled vehicles are springs, shock absorbers, bogies, axles, wheels, and tires. Not all of these components are necessary for a suspension system. In some cases, the particular component(s) may be modified to perform the function of one or several of the other components. Suspension components suitable for wheeled amphibians are essentially the same as for any military wheeled vehicle and are discussed in another Handbook, Ref. 26.

7-8.1 CRITERIA FOR SUSPENSION SYSTEMS

Criteria which must be considered in the design of the suspension system include maintainability, simplicity, durability, reliability, vulnerability, mobility, weight, and, of course, economy. Most of these qualities are defined and discussed in par. 5-1.3. In particular, however, mobility was not included as a criterion for design. In the present context, the term mobility is limited to apply to the ground mobility of the amphibian.

In Ref. 63, mobility is defined as "the competence of a vehicle to perform its mission as measured by its best average speed over a route representative of the terrain where it will operate". The average speed of a vehicle in a given terrain is the proper measure of its mobility for that particular terrain. Thus, an amphibian's mobility might be categorized as soft-soil mobility, sand mobility, etc., or could be based on some particular mission profile. The average speed should be based on the straight line distance between two points regardless of the actual path that the vehicle must travel in order to negotiate the terrain.

7-8.2 GENERAL DISCUSSION OF DESIGN

The strength and dynamic response characteristics of the suspension system components will be designed to withstand the forces (including shock loadings) and to provide the flexibility necessary to isolate the operating personnel from excessive motions and vibrations. Discussion of these forces and motions is presented in par. 7-3.1. More detailed information can be found in Refs. 26, 63, and 120. Design variables include the ratio of unsprung to sprung mass, spring constants, and damping coefficients.

Although the suspension system is primarily an impact absorbing mechanism, its design is influenced greatly by the requirements for power transmission, mobility, and steering. These requirements dictate that all the wheels be powered, hence "live" axle assemblies must be used to drive the wheels. It is also probable that all wheels will be steerable (par. 7-9.4) and that the system will use "independent suspension" in which each wheel is free to oscillate independently of the others. The requirements for marine operation also influence the suspension system design.

7-8.3 EFFECTS OF MARINE ENVIRONMENT

Amphibian waterborne operation requirements affect the design of the suspension (aside from the obvious problems of corrosion and electrolytic action). Most recent amphibian designs have placed increased emphasis on waterborne performance with the result that "unsprung" or rigid suspension systems have been favored. While waterborne, the suspension system serves no useful purpose (with the possible exception of steering by means of the wheels, par. 7-9.3) and only adds to the displacement and drag of the amphibian. It has been estimated that 65 percent of the total water resistance of the DUKW is attributable to the exposed suspension system and its associated wheel-well cutouts. Thus, in order to minimize the weight and drag of the suspension system, springs and shock absorbers have been eliminated, sole reliance being placed in the tire design to provide adequate flexibility and damping in the system.

The LARC V is a good example of an amphibian having a rigid suspension system. While its top water speed is undoubtedly higher than could be achieved by a similar amphibian with a conventionally sprung suspension, its top land speed is decidedly lower (25 mph compared to about 50 mph).

As long as primary interest is focused on the waterborne performance of amphibians, it is reasonable to assume that unsprung suspension systems will prevail.

7-8.4 VEHICLE-SOIL MECHANICS

7-8.4.1 Shearing Strength of Soil

The shearing stress τ which a soil can support is given by the classic Coulomb's equation

$$\tau = c + \sigma \tan \varphi \quad (7\text{-}129)$$

where

c = coefficient of cohesion, psi
σ = normal stress on the soil, psi
φ = angle of friction, deg

Dry sand has no cohesion ($c = 0$), whereas in wet clay there is no friction ($\varphi = 0$). Most actual soils are neither purely cohesive nor purely frictional. The important point to remember is that traction in different types of soil is supported by entirely different mechanisms (cohesion and friction) and, thus, mobility in different soils depends upon different parameters. It follows that an amphibian with good (relative) sand mobility does not necessarily have good mobility in clay soils.

7-8.4.2 The Cone Index

The cone penetrometer was developed by the Corps of Engineers to obtain a simple, straightforward measurement of soil "trafficability". It is one of the most widely used instruments for determining an index of the soil strength in the field. The cone penetrometer's value as a tool for measuring bearing and tractive capacity lies in the fact that a vast number of tests have been performed with it to correlate vehicle performance with soil condition.

The cone penetrometer is a right, circular, 30-deg cone with a base area of 0.50 in.^{2} The *cone index* (psi) is the force that must be applied to the penetrometer per square inch of cone end area to force it into the undisturbed ground. The force required is measured at increments of depth so that the gradient of the cone index can be computed. The *rating cone index* is similar to the cone index but is determined after the same soil has been pounded in a mold with 100 blows of a 2½-lb tamper falling 12 in. The ratio of the rating cone index to the cone index (i.e., the ratio of the index after and before tamping) is called the remolding index (dimensionless) and can be greater or less than 1.0. The rating cone index is an overall or final measure of the soil's ability to support sustained traffic. A *vehicle cone index (VCI)* is the minimum value of the rating cone index at which a particular vehicle can negotiate the soil.

The cone index has many shortcomings as a quantitative index of mobility. It has been correlated only empirically with the "go and no-go" soil conditions with respect to particular existing vehicles in a limited number of ground types. While the method is expedient, it does not account for all soil conditions; it is not too accurate; and it does not permit prediction of the performance of nonexistent, unorthodox vehicles in the design stage.

7-8.4.3 The LLL (Land Locomotion Laboratory) Soil Value System

The LLL soil value system (Ref. 93) is a potential design tool to estimate the soft soil mobility of new wheeled amphibians. This system takes into account more of the soil parameters than the vehicle cone index, and incorporates allowances for a number of detailed types of soil-vehicle behavior. Two sets of independent soil parameters are used: (1) the direct shear reactions c and φ (Eq. 7-129) measured by plate penetration tests and a normally loaded annular shear vane, respectively, and (2) the soil sinkage parameters K and n.

$$p = Kz^n, \text{ psi} \quad (7\text{-}130)$$

where

p = unit ground pressure, psi
K = modulus of deformation
z = sinkage, in.
n = coefficient of sinkage, 0.5 for sandy loam

K and n are experimental coefficients for specific soil structures. K is a function of the form and size of the loading area as well as the soil physics. If K is expressed in terms of the "cohesive" modulus K_c and the frictional modulus K_φ, the parameter is made "independent" of size (Ref. 94).

$$K = \frac{K_c}{b} + K_\varphi \quad (7\text{-}131)$$

where b = breadth of footing.

For a complete discussion of the refinements and use of LLL soil value system, refer to Ref. 93. The main problem with this system is that it is not fully verified experimentally.

7-8.4.4 Correlation of Various Soft Soil Indexes

There are other indexes in addition to the vehicle cone index and LLL soil value system

which have been developed to predict relative performance in soft soils, Refs. 26, and 95 to 100. Nuttall, Ref. 101, has attempted to correlate the theories and test results of the various agencies pertaining to soft-soil performance in different soil types. When certain simplifying assumptions are made, one basic parameter—the nominal unit ground pressure (*NUGP*)—is found to be common to all the mobility indexes. The *NUGP* is arbitrarily defined as

$$NUGP = \frac{w}{br}\text{, psi} \qquad (7\text{-}132)$$

where

w = average gross load carried on a single tire, lb

b = undeflected tire section width, in.

r = undeflected tire radius, in.

The Army Tank-Automotive Command (ATAC) uses a similar expression to calculate the average ground pressure

$$p_{av} = \frac{w}{0.7rb}\text{, psi} \qquad (7\text{-}133)$$

Thus, p_{av} is related to *NUGP* by the simple constant 0.7 which is introduced as a "shape factor" to account for the elliptical shape of the theoretical footprint area.

A summary of Nuttall's findings is given in Table 7-44. These results show that, in clay soils, reduction of *NUGP* of the tire will clearly extend the operating range into weaker soils. In other words, an amphibian with lower nominal unit ground pressure (less weight or larger, broader tires) will go in soils with less cohesion. In sandy soils, the *NUGP* is, again, a controlling factor. However, larger vehicles (or larger tires) may operate at high *NUGP* without loss in performance relative to smaller vehicles in sand.

Table 7-45 from Ref. 101 presents the critical soft soil mobility indexes for various wheeled amphibians along with their payload and top speeds in water and on land. The indexes are calculated from the original equations of the various agencies (not the equations in Table 7-44). The LARC V has the best relative soft soil mobility of the amphibians presented in the table.

Fig. 7-96 has been included in order to illustrate the relationship between a soft soil mobility index and actual mobility on a "go" or "no-go" basis. This figure must be considered as preliminary; however, more complete data would exhibit the same characteristics. A few examples will illustrate the use of this figure:

(1) The 50-pass vehicle cone indexes for the LARC V and DUKW (from Table 7-45) are given as VCI = 53 and 63, respectively. From Fig. 7-96 for the tropical climate during the wet season the frequency of occurrence of soils with rating cone index below the critical value for each vehicle is estimated to be 21 and 27 percent, respectively. Conversely, the percentage of all land area which is trafficable is 79 for the LARC and 73 for the DUKW. Thus, it is estimated that the LARC is capable of negotiating about 22 percent of the area which is impassable for the DUKW or 6 percent more of the total area.

(2) Comparing the single pass mobility of the LARC V and LARC LX (VCI_1 = 34 and 70, respectively) for the temperate climate and high-moisture condition, the respective trafficable areas are estimated to be 82 and 40 percent of the total area, i.e., less than half of this land area open to the LARC V (on a single pass basis) can be negotiated by the LARC LX.

7-8.4.5 Other Aspects of Terrain-vehicle Technology

7-8.4.5.1 Roughness

The dynamic behavior of an amphibian operating at any speed over hard ground with known geometry (roughness) can be treated statistically with methods analogous to those used to predict motions and resistance in waves (par. 7-4.1.2.4). The important parameters are:

a. The geometry of the surface

b. The geometry of the amphibian

c. The mass of the amphibian and its distribution

d. Articulation (if any)

e. Elastic and damping characteristics of the suspension system components

f. Coefficients of traction of the running gear on elements of the surface

The accuracy of motion prediction depends entirely upon the accuracy of the dynamic constants and the detail with which the terrain-vehicle system is mathematically modeled. The vibrational motions and accelerations of a particular amphibian are found as relatively simple functions of speed and the power spectrum density (PSD) of the land

TABLE 7-44 COMPARISON OF SOFT SOIL MOBILITY INDEXES

Soil Type	Basic Source	Critical Index	Units
Clays	1 FVRDE rigid wheels with sidewall traction	c = the larger of $\begin{cases} 0.25\ NUGP\left(1+\sqrt{1+\frac{D}{4b}}\right) \\ \text{or} \\ 0.264\ NUGP \end{cases}$	psi
	2 FVRDE rigid wheels w/o sidewall fraction	c = 0.50 NUGP	psi
	3 LLL rigid wheels w/o sidewall traction	$\begin{cases} n = 0;\ c = 0.50\ NUGP \\ n = 0.5;\ c = 0.40\ NUGP \end{cases}$	psi
	4 WES single tires in the laboratory	$\overline{CI}$ = 2.5 NUGP	psi
Fine Grained Soils (loams, silts,clays, etc.)	5 WES 50-pass (from	VCI = 4 NUGP + 14	psi
	6 WES single pass (from field tests)	VCI = 3 NUGP + 3	psi
A Sandy Loam	7 LLL rigid wheels w/o sidewall traction	K = NUGP $(2/d^{0.5})$ (NUGP > 4.85 psi)	$lb/in.^{2.5}$
"Optimum"	8 EKLUND	M_e' = NUGP $(2.2/d^{0.85})$	nd
Sands	9 WES field tests	G_1 = NUGP $(6/b^{0.5}d^{0.5})$	psi/in.
	10 WES single tires in laboratory	G_4 = NUGP $(8/b^{0.5}d^{0.5})$	psi/in.

NOTES: FVRDE - British Laboratory, Ref. 95
LLL - Land Locomotive Laboratory, Ref. 93
WES - U. S. Army Engineer Waterways Experiment Station, Refs. 96 to 99
EKLUND, Ref. 100
NUGP = w/br, psi

where w = average load on single tire, lb

b = undeflected tire width, in.

r = d/2 = undeflected tire radius, in.

D = undeflected tire diameter, in.

The critical index is the minimum value of the index at which a vehicle will just maintain headway in unaccelerated, level, straight-line operation.

c = coefficient of cohesion of soil, psi

$\overline{CI}$ = average cone index of soil, psi

VCI = average rating cone index of field soils for a minimum of 50 passes in the same ruts, psi

VCI_1 = average rating cone index of field soils for single pass, psi

K = soil consistency in selected sandy loam where n = 0.5, c = 0.56 and tan φ = 0.31

n = an exponent expressing the sinkage characteristics of the soil

φ = the angle of friction (Eq. 7-129)

M_e' = the inverse of the Eklund Mobility Factor (not in percent). M_e' greater than 100 indicates an overloaded tire and less than optimum off-road performance

G = average gradient to cone index of sand versus depth in 0.6 in. layer

TABLE 7-45 PERFORMANCE CHARACTERISTICS OF SEVERAL WHEELED AMPHIBIANS

Amphibian	Soft Soil Mobility Indexes						Pay-load tons,	Speed, mph	
	VCI	VCI_1	G_{sand}	M'_e	K_{loam}	c_{clay}		Land	Water
1943 DUKW	63	43	4.6	101	4.2	5.1	2.5	50	6.5
1951 SUPERDUCK	82	53	5.3	137	5.0	6.4	4.0	48	6.7
DRAKE	92	56	5.0	125	5.0	6.7	8.0	45	9.0
1960 LARC V(u)	53	34	2.4	85	2.6	4.0	5.0	25	8.6
LARC XV(u)	82	54	3.4	96	3.8	6.5	15.0	20	9.5
LARC LX(u)	103	70	3.2	102	4.2	8.6	60.0	14	7.0
High-speed Amphibians									
1962 LVW-X1(u)	80	53	4.2	103	4.3	6.2	5.0	35	35
1964 LVH-X1	40	81	4.1	104	4.3	6.3	5.0	40	40
Truck									
1960 M34	62	37	3.9	102	3.4	4.4	2.5	—	N.A.

Notes: A lower value of the index indicates greater relative mobility (except M'_e).

(u) = unsprung suspension.
$N.A.$ = not applicable
VCI = average rating cone index of field soils for a minimum of 50 passes in the same ruts, psi
VCI_1 = average rating cone index of field soils for a single pass, psi
G_{sand} = average gradient to cone index of sand vs depth in 0.6 in. layer
M'_e = inverse of Eklund Mobility factor (not in percent)
K_{loam} = soil consistency in sandy loam
c_{clay} = coefficient of cohesion in clay soils, psi

surface, Ref. 102. The human responses to the motions are discussed in par. 9-2.1.2.

7-8.4.5.2 Obstacles

Obstacles, geometric features too large to be considered as roughness, include such things as boulders, trees, stumps, and ditches (see Ref. 103). The parameters which affect an amphibian's potential ability to deal with obstacles are mainly geometrical including:

a. Wheelbase
b. Wheel diameter
c. Approach, departure, and break angles
d. Ground clearance (both static and dynamic)
e. Weight distribution
f. Suspension compliance with features of the terrain

Obstacle crossing is a "go" or "no-go" situation for the most part; speed and vibrations are of secondary importance.

7-8.4.5.3 Bank Egress

The bank egress capability may be considered to be a combination of the soil mechanics and obstacle problems. Stream and river banks are characterized by steep slopes and high, vertical steps. The steep slopes are often wet and slippery; where the slopes are more favorable, the soil is likely to be composed of weak, waterlogged silts as in marshes and bogs. Thus, the problem of bank egress severely limits an amphibian's riverine mobility. It is safe to say that no existing amphibian is capable of crossing a majority of the larger streams and rivers in this country if the point of egress is chosen at random.

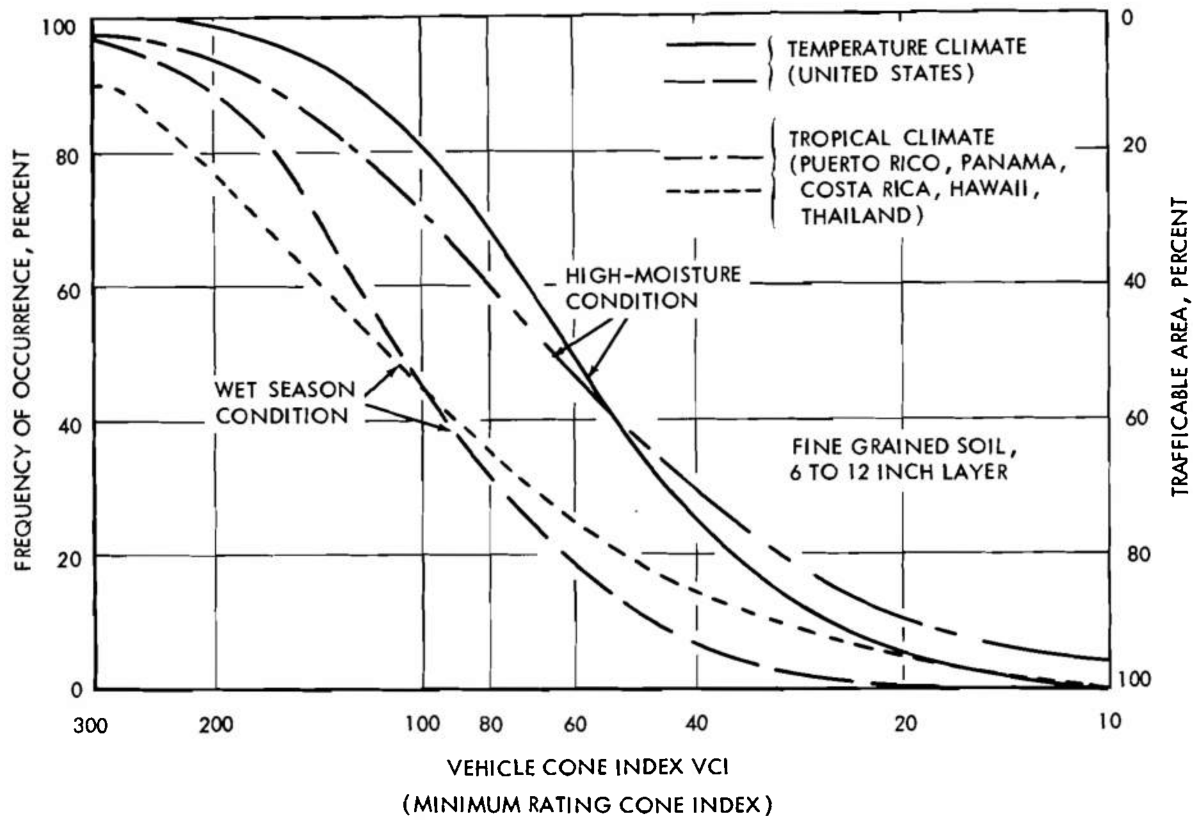


Figure 7-96. Cumulative Frequency of Rating Cone Indexes

7-8.5 TIRE SELECTION

In a rigid suspension system, the pneumatic tire must perform the functions of both a spring and a shock absorber (damping) as well as its normal role of providing traction for propulsion. The spring rate k of a tire is a complicated function of compression of the air in the tire and the flexure of the sidewalls. The spring rate is not a constant, but has definite nonlinearities characterized by an increase in k as deflection begins and then a decrease in k with increased deflection. There is also some damping inherent in the flexure of the tire; however, it is believed that the greater portion of damping with respect to tires is between the tire and the soil surface and within the soil itself (Ref. 120).

The damping in the tires is insufficient with respect to operations on hard surfaced roads. The LARC V land speed is limited to about 25 mph. Any attempt to increase the speed results in large amplitude motions (galloping) at the system's resonant frequency because of underdamping. With suitable damping, the amphibian would be capable of road speeds on the order of 50 mph. Inadequate damping must be accepted in a rigid suspension system unless some future development in tire design results in a major improvement in tire damping without seriously degrading rolling resistance.

7-8.5.1 Tire Tread

The choice of tire tread will be a compromise, in most cases, between conflicting requirements for different soil conditions. For sand, hard soils and roads, a shallow-ribbed pattern is desired. A deeper, lugged tread is needed in soft platic soils where the tires must excavate soil in order to get down to the hardpan. A deeper tread, of course, contributes greatly to the rolling resistance of the tire.

7-8.5.2 Estimating Tire Weight (Ref. 101)

An empirical fit to published data on the weight W_t of a number of existing tires leads to the following equation:

$$W_t = K_1\left(\frac{bd^2 W_1^{1/2}}{d_r}\right) + K_2\text{, lb} \quad (7\text{-}134)$$

where

W_l = maximum rated load on the tire at its designed inflation pressure, lb. If highway operation is included, the load (and, implicitly, the inflation pressure) for this service is used.

b = undeflected tire section width, in.

d = undeflected tire outside diameter, in.

d_r = rim diameter, in.

K_1 and K_2 = constants which vary somewhat for various types of tires. Average values are tabulated below:

		K_1	K_2
Highway	Mileage Tread, standard	0.0015	10
	Duplex	0.0011	20
	Extra Small	0.0017	1
Off-Road	Grader	0.0014	20
	NDCC* Military(1)	0.0013	0
	Sand	0.0009	20
	Earthmover	0.0011	100
	Rock Tread	0.0015	50
	Duplex, Rock Tread	0.0011	20
	Compactor (slow speed)	0.0011	0
Terra-Tires (2)	Smooth	0.0005	5
	Terragrip Tread	0.0006	30

(1) Casing only: Tubes and flaps add approximately 75 percent.

(2) The scatter of actual Terra-Tire weights versus values computed by this equation is relatively great, suggesting that the form of the equation is not entirely appropriate for these extreme types—which is not surprising. The constants given provide rough guidance, nonetheless.

*NDCC Nondirectional Cross Country

SECTION IV STEERING SYSTEM DESIGN

7-9 STEERING SYSTEM DESIGN

7-9.1 WATERBORNE STEERING THEORY

7-9.1.1 Steering Forces

The forces on a rudder follow the same principles as forces on low aspect ratio airfoils and hydrofoils. They are usually expressed as dimensionless lift, drag, and moment coefficients denoted by C_L, C_D, and C_M, respectively. Lift is defined as the force normal to the direction of inflow and drag is parallel to the inflow. Rudder lift is directed in the horizontal plane and normal or nearly normal to the amphibian centerline. The relationship between the forces and their coefficients is as follows:

$$\left.\begin{aligned} L &= C_L\left(\frac{\rho}{2}\right) Sv^2\text{, lb} \\ D &= C_D\left(\frac{\rho}{2}\right) Sv^2\text{, lb} \\ M &= C_M\left(\frac{\rho}{2}\right) Scv^2\text{, ft-lb} \end{aligned}\right\} \quad (7\text{-}135)$$

where

L = lift force, lb

D = drag force, lb

M = moment about any reference axis, ft-lb

S = rudder area, ft^2

v = mean flow velocity over rudder, ft/sec

c = chord of rudder, ft

ρ = density of water, slug/ft^3

The force and moment coefficients are functions of angle of attack α, aspect ratio A, section shape, and planform. Determination of these coefficients are discussed in par. 7-9.2.2.

The forces generated for any particular rudder are mainly determined by the effective angle of attack α of the rudder and the relative water speed v. In open water, α and v are readily determined, but behind a ship the conditions are not as simple. Fig. 7-97 illustrates these conditions. In this figure, α is the effective angle of attack, δ is the rudder helm angle, ϕ is the drift angle of the ship, and ϵ is due to the straightening influence of the hull, appendages, and propeller. The velocity relative to the rudder is a function of the ship speed, hull wake, and propeller slipstream.

The turning moment applied to the ship by the rudder is given by

$$M = [L\cos(\delta - \alpha) - D\sin(\delta - \alpha)]\ell + M_r\text{, ft-lb} \quad (7\text{-}136)$$

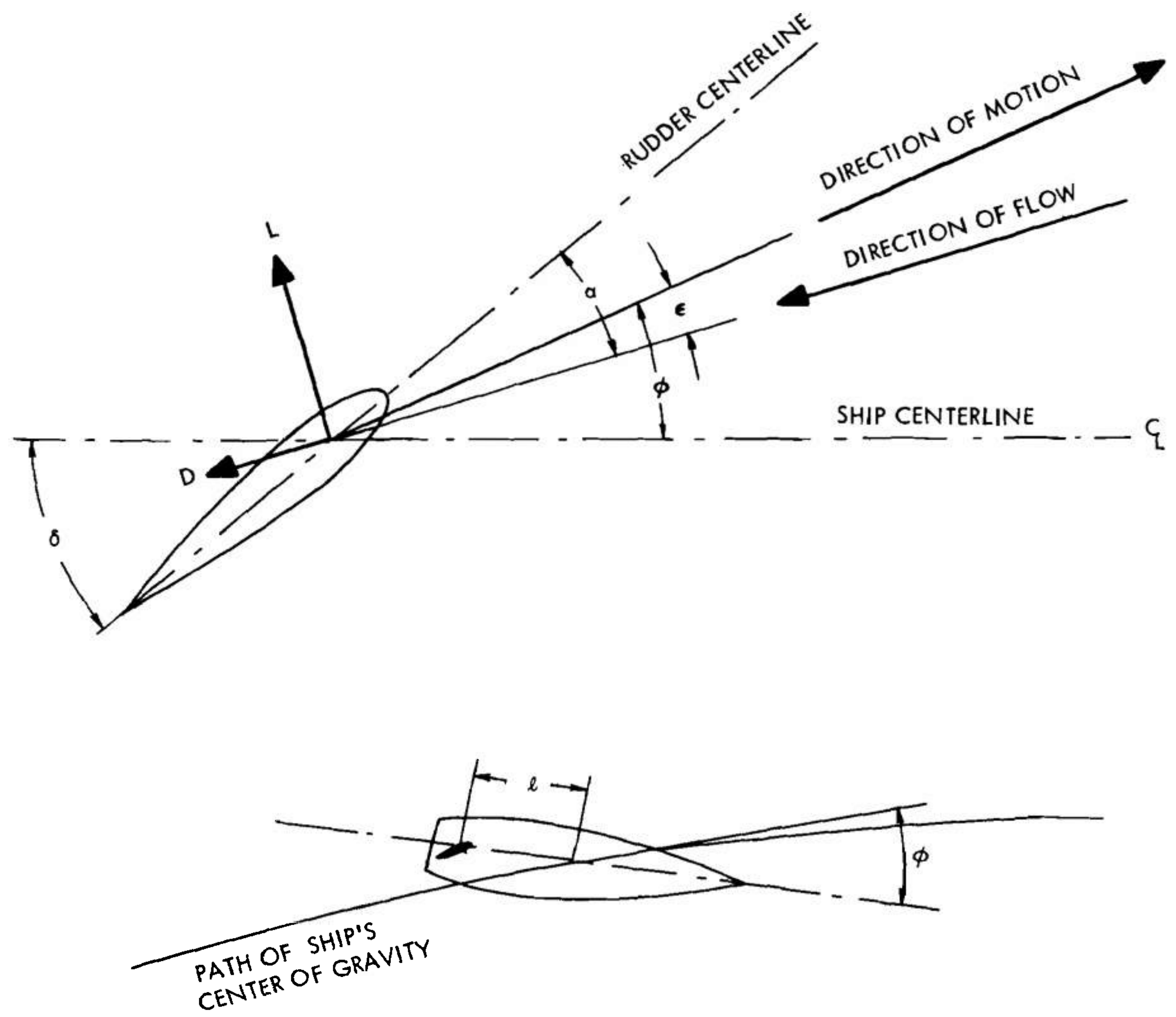


Figure 7-97. Effective Rudder Angle for a Rudder Behind a Ship During a Turn

where

M = moment about the ship's CG, ft-lb
M_r = moment about the rudder stock, ft-lb
L = rudder lift force, lb
ℓ = arm from rudder stock to ship's CG, ft
a = effective inflow angle, deg
δ = rudder helm angle, deg

In a maneuver, the angles a and δ will change from the instant the rudder is put over until a new state of equilibrium is reached. It takes some time to put the rudder over to the desired helm angle, but this time is short enough that the ship's heading has not changed very much when the desired angle is reached. At this instant then, the angles ϕ and ϵ are negligible so that the angle of attack is equal to the rudder helm angle ($a = \delta$) and Eq. 7-136 reduces to:

$$M_i = L\ell + M_r \tag{7-137}$$

where

M_i = initial turning moment applied to the ship by the rudder

The initial moment to turn the ship can be made larger by increasing the lift on the rudder and increasing the moment arm by placing the rudder far behind the center of gravity. The moment about the rudder stock M_r is usually negligible. The lift is directly proportional to the lift coefficient C_L, area S, and the square of the inflow velocity v (see Eq. 7-135). The lift coefficient is a function of the angle of attack and is maximum when the rudder is at the "stall angle".

Once the ship starts turning, hydrodynamic forces are generated by the cross-flow over the hull. This flow is comprised of velocity components due to the drift angle (sway) and angular velocity (yaw) of the ship. On a directionally unstable ship, these hydrodynamic forces create a moment in the same direction as the rudder moment and tend to decrease the turning radius. At the same time the drag on the ship is increased causing a speed reduction. With the rudder helm angle unchanged, the ship will continue to tighten its turn until a steady turning motion is reached. At this equilibrium condition, the hydrodynamic moments on the ship and the rudder are balanced (equal and opposite).

Fig. 7-98 is a schematic diagram of a ship executing a steady turn maneuver. This maneuver is useful in determining some of the parameters which are used to measure a ship's relative maneuverability. The steady turning radius, advance at 90° change in heading, transfer at 90° change in heading, and tactical diameter are often made dimensionless by dividing by the ship's length. For example, the tactical diameter of the LARC V with a rudder angle of 45° is about 2.5 ship lengths with an approach speed of 9 mph.

The ship's inertia forces have not been included in this discussion. The complete equations of motion are presented in Ref. 67.

The moment on the rudder stock M_r is important in determining the strength of the steering gear and linkages. In addition to the static forces, however, consideration must be given to the additional torque needed to attain the desired angular rate of rudder deflection. The product of angular rate and the maximum torque determines the power which must be supplied by the steering gear (or helmsman). The rudder torque in many cases is greatest when the ship is going astern. If so, this condition will determine the strength requirements.

Steering forces can be produced without resorting to rudders. In many cases, the alternatives provide a greater degree of maneuverability and control. Vertical axis propellers, water jets, and right angle drives are

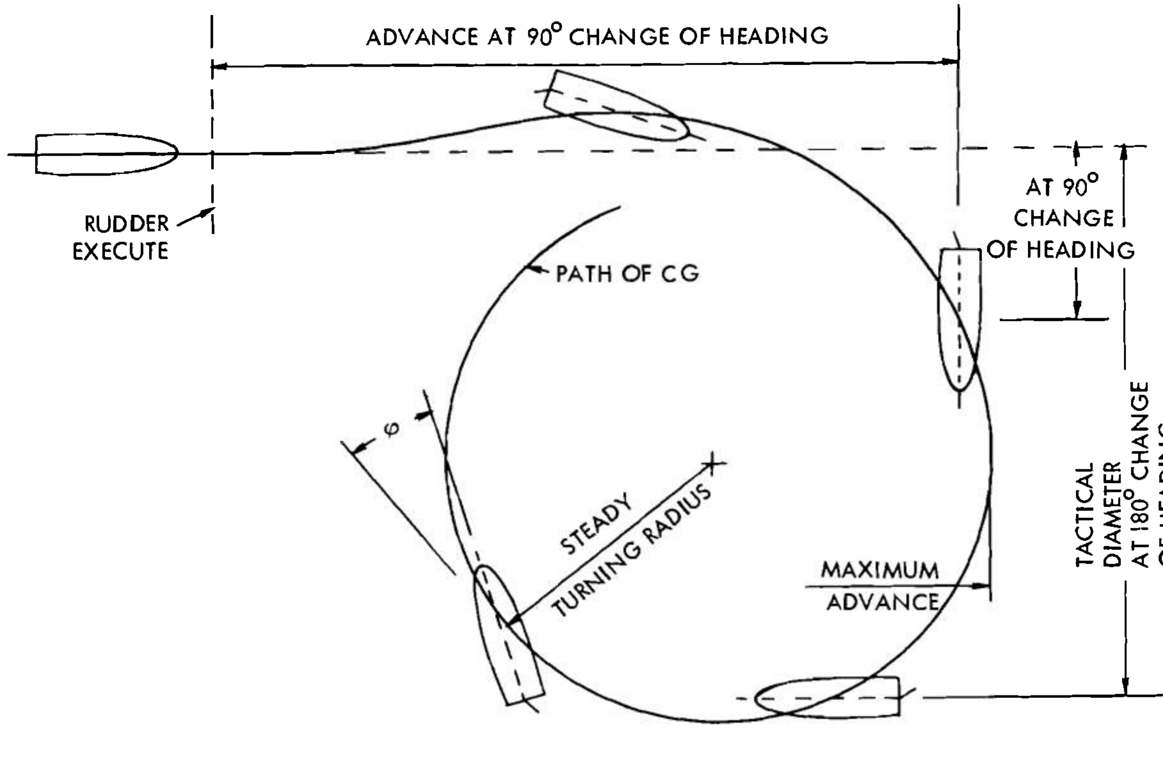


Figure 7-98. Geometry of a Turning Circle

examples in which the thrust of the main propulsor can be directed in any desired direction to produce a turning moment about the ship's center of gravity. Perhaps the greatest advantage that these devices have over rudders is that large turning moments can be generated at low and zero advance speeds.

The active rudder consists of a small propeller driven by a submersible electric motor which is housed in a nacelle on the rudder. At high speeds the active rudder is used as a conventional rudder. But at low speeds the propeller is activated to produce thrust which is trainable by moving the rudder in the desired direction.

The steering nozzle is simply a Kort nozzle which can be rotated about a vertical axis through the plane of the propeller. This "tube" rudder produces superior turning forces by diverting the entire propeller slipstream while most of the advantages of a Kort nozzle for propulsion are retained. The primary disadvantage lies in the nozzle's vulnerability to damage.

Ducted thrusters are finding increased application on merchant vessels. A transverse tube open to the sea is located deep in the bow of the ship and houses a propeller which may have controllable pitch and is likely to be driven by an electric motor. At slow speeds, activating the bow thruster gives lateral thrust and large turning moments. The ducted thruster becomes less effective at higher speeds and conventional rudders are used for high speed maneuvers. It does not seem very likely that ducted thrusters will be used on future amphibian designs because of problems with arrangements, lost buoyancy, and increased complexity.

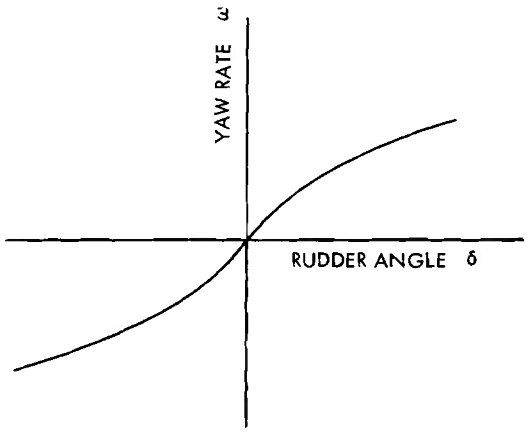
Many other devices for producing steering forces are conceivable, but one application which is worthy of particular note concerning wheeled amphibians is the use of the wheels as crude rudders. It is essential that steering of the wheels of an amphibian be coordinated with the rudder when operating in surf so that a smooth transition may be made from water to land operation. The front wheels are quite effective as bow rudders and invaluable for control in a following sea (par. 7-9.1.5). Indeed, the tactical turning diameter of a recent high-speed amphibian in the surf mode of operation was not appreciably changed when the rudder was disconnected and steering was effected with the wheels alone.

7-9.1.2 Directional Stability and Maneuverability

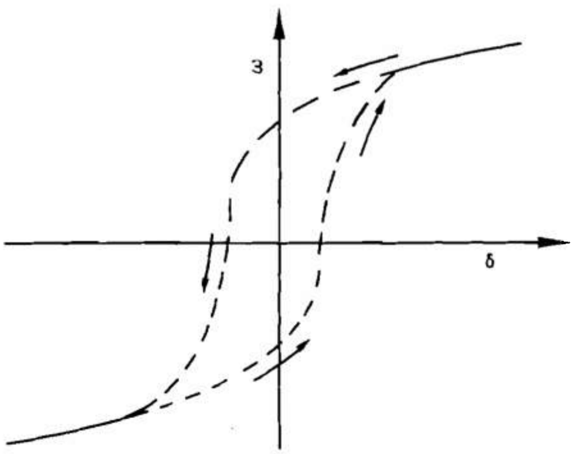
Maneuverability may be defined as a ship's ability to follow a desired course as steered by the helmsman. The degree of maneuverability depends on the rapidity and magnitude of response to the helm and on a consistent, one-to-one relation between rudder angle and steady turning rate. Directional stability implies that this one-to-one relation exists. Unfortunately, a high degree of directional stability in a ship is associated with slow turning rates and large tactical diameters, i.e., limited maneuverability.

The degree of directional stability or instability is commonly determined for a free running scale model by the spiral maneuver. In this test, the rudder is put over in a succession of discrete angle increments to one side, to the other side, and back to zero. Each rudder angle is maintained until a steady yaw rate is established and recorded. The results are plotted with steady yaw rate ω as a function of rudder angle δ as in Fig. 7-99. A stable ship has a single curve through zero rudder angle. The degree of stability is qualitatively indicated by the slope of the curve at the origin; the lower the slope, the higher the stability. However, for a given ship the slope of this curve is a function of the rudder lift-curve slope. Hence, a comparison of stability between two ships should be limited to ships having similar rudder effectiveness. A hysteresis-type loop, Dieudonne loop, around the origin is characteristic of unstable ships. The existence of a Dieudonne loop signifies that continuous rudder activity will be required to maintain a steady course. The size of the loop gives a qualitative measure of instability. Its height indicates the extent to which the turning rate will vary with the rudder held amidships, i.e., its instability. The width of the loop is a fairly good measure of the ship's course keeping ability because it defines the envelope of rudder angles which must be used to hold a course.

The directional stability of a ship depends upon its lines or hull form and its appendages such as skegs or deadwood. The size of the amphibian's appendages should be kept small unless the amphibian is directionally unstable, in



(A) DIRECTIONALLY STABLE SHIP



(B) DIRECTIONALLY UNSTABLE SHIP

Figure 7-99. Results of Spiral Maneuvers

which case appendages may be enlarged or skegs added to attain stability. It is particularly important that amphibians be directionally stable so that response to the helm is positive and rapid for difficult surf operations. As long as the amphibian is stable, any increase in the degree of stability is undesirable because it leads to slower response and increased turning diameters unless larger more effective rudders are employed.

7-9.1.3 Steering at Very Low or Zero Speed of Advance

The force producing capabilities of a rudder are proportional to the square of the velocity of flow over its surface. As ship speed decreases, the effectiveness of the control surfaces also decreases. At zero speed of advance, any rudder forces are limited to those which arise from velocities in the slipstream of a thrusting propeller. In this case the turning force comes from the change in momentum caused by the rudder deflecting the flow in the propeller race. The lift on the rudder will be a rather small fraction of the propeller thrust. The thrust of the propulsor is more effectively used for low speed maneuvering if it can be trained in any desired direction as with vertical axis propellers, vertical shaft right angle drives, and water jets. With these devices the thrust may be directed at right angles to the centerline so that a turn may be executed without imparting any forward motion to the vessel.

In many amphibian supply operations it is necessary to hold the amphibian next to a large ship for cargo transfer. If a springline is attached to the bow of the amphibian, its rudder may be used to hold the stern against the cargo ship while maintaining propeller thrust against the spring line. With a trainable propulsor, an amphibian may maintain a station next to the supply ship without passing a line between the hulls.

7-9.1.4 Steering When Reversing

The forces supplied by a rudder for steering at low speeds are smaller when the propeller is reversed to apply astern thrust. Inflow velocities to a propeller are small compared to the outflow in the race so that when going astern the flow velocity past the rudder is relatively low. River towboats are often equipped with flanking rudders, ahead of the propeller, which are operated only when the propeller is reversed, at which time they are effective for deflecting the propeller race to produce steering forces.

While the speeds and rudder forces in astern operations are generally low, the torque on the rudder stock may be maximum for this condition. The axis of rotation for a balanced rudder will pass through or near the quarter-chord point of a section. In ahead operations, the center of pressure for a properly designed balanced rudder will be within 0.05 chord length of the axis. However, in astern operation the center of pressure will be at about the 3/4-chord point or about 0.5 (chord) behind

the axis. Thus, the moment arm of the rudder lift about the rudder stock is about ten times larger for astern operation. The maximum torque, of course, is the product of the lift times its respective moment arm.

7-9.1.5 Steering in Rough Water and in Surf

The steering ability of any ship is adversely affected by waves. The worst condition occurs when the ship is running in a following sea with an advance speed about equal to the wave celerity. This is because the hydrodynamic yaw forces created by following waves are applied over a longer period of time so that the resulting yaw motion is more severe.

Amphibians are required to operate safely through the surf. Waves always become higher and steeper as they approach a beach from open water. Thus, the need for a high degree of maneuverability is most pronounced for an amphibian landing through surf. The greatest pitfall facing the helmsman is the tendency of the amphibian to broach in the surf. He must quickly apply opposite rudder to counteract any tendency of the amphibian to veer off course and the rudder (or other steering device) must be effective in creating a turning moment to resist the yawing motion. If the vessel should broach near the beach line, it is likely that it will subsequently be swamped or overturned.

An amphibian that is directionally stable in calm water may behave as if unstable when it is "riding" a wave through the surf. This behavior is attributable to the orbital velocities of the water particles in the waves, as illustrated in Fig. 7-100. The arrows in this figure are vectors which indicate the direction and velocity of particle motion in the fluid. An amphibian is shown poised on the wave front with its stern at the crest of the wave and bow in the preceding trough. The water velocity relative to the amphibian is nearly zero at the stern and about twice the amphibian's forward speed at the bow. Thus, the stabilizing influence on any appendages at the stern of the vessel is lost because there is no flow there, whereas the yaw forces due to asymmetry of flow past the bow (from some initial yaw perturbation) are larger than normal with the high relative velocities in the bow.

Steering by means of a conventional rudder in the foregoing example is also handicapped by the low relative flow velocity past the stern due to the orbital trajectories in the waves. On the other hand, steering by means of the front wheels is enhanced by the high relative velocities encountered at the bow of amphibian.

7-9.2 DESIGN OF MARINE STEERING SYSTEMS

7-9.2.1 Design Criteria

The main objective of the requirements for the marine steering system is to insure that the amphibian will be able to safely negotiate some prescribed minimum surf. Successful operation through the surf zone depends largely on experience and good seamanship. However, the characteristics of the amphibian with respect to roll stability and maneuverability are also important. Two criteria used to judge maneuverability are directional stability and tactical turning diameter. These qualities are defined and measured only for calm water maneuvering. The tacit assumption is made that an amphibian with good maneuverability in calm water will also exhibit good maneuverability in rough water. In general this assumption is probably well founded. However, for the particular case of an amphibian in following seas, conventional rudder control may be compromised so that some alternative steering device may be desired. The low speed and zero speed maneuverability of an amphibian design may be required to meet some specification so that even in rough water it will be able to reach and maintain a station next to a ship for cargo transfer.

The previous discussion applies to the performance of the amphibian, which is only one of the many design criteria which will be used to evaluate the steering system design. Other criteria are described in par. 5-1.3 and include maintenance, simplicity, ruggedness, vulnerability, reliability, weight, arrangements, and economy.

Much of the time it will not be sufficient to apply the design criteria to the steering system as an entity. For example, the performance of the steering system is reflected in the amphibian's degree of maneuverability. But maneuverability is a function of the hull design and its appendages as well as the steering system effectiveness. Thus, if a change in the steering system also alters the hull in any way, then the

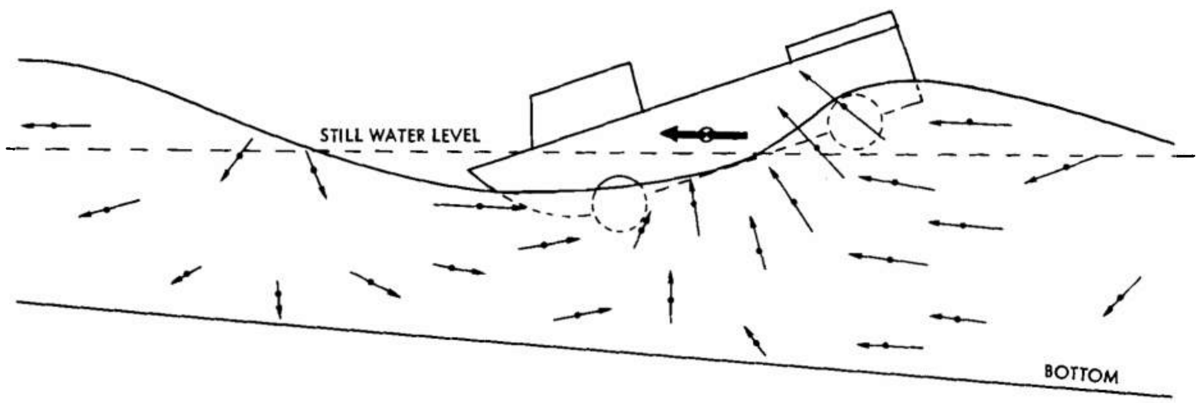


Figure 7-100. Schematic Diagram of a Wave Approaching a Beach Showing the Distribution of Particle Velocities

steering performance and all the other criteria, as well, must be applied to the hull and steering system together. Unfortunately, there are many interrelationships among the various systems on an amphibian design so that evolution of the design based on all the criteria becomes quite complex and difficult.

7-9.2.2 Rudders

7-9.2.2.1 Theory of Operation

The main purpose of the rudder is to produce moments about the center of gravity of a ship with which to initiate and control turning motions. It is important to know the lift of the rudder at a given speed and rudder angle and, especially, the maximum lift force which can be produced by the rudder. The turning moment is a function of rudder lift L as in Eqs. 7-136 and 7-137. The rudder lift is a function of velocity and the lift coefficient C_L (Eq. 7-135). The lift coefficient is, in turn, a function of rudder aspect ratio and angle of attack. The maximum value of lift coefficient is a function of Reynolds number, aspect ratio, and thickness-chord ratio.

The lift coefficient is approximately a linear function of angle of attack up to the stall angle. Thus, the lift coefficient C_L at any angle below the stall angle can be approximated from the lift coefficient slope dC_L/da:

$$C_L = a(dC_L/da) \tag{7-138}$$

where a is the rudder angle, deg.

Fig. 7-101, taken from Ref. 104, shows an empirical curve of lift coefficient slope as a function of aspect ratio AR for rectangular lifting surfaces. Aspect ratio is the ratio of span to chord, thus

$$AR = \frac{b^2}{S} \tag{7-139}$$

where

AR = aspect ratio of the foil, nd
b = span of the foil, ft
S = area of the foil, ft^2

The lift coefficient slope is relatively independent of the thickness of the foil.

The maximum lift coefficient for symmetrical foils with thickness-chord ratio t/c = 0.15 is shown in Fig. 7-102 taken from Ref. 105. The

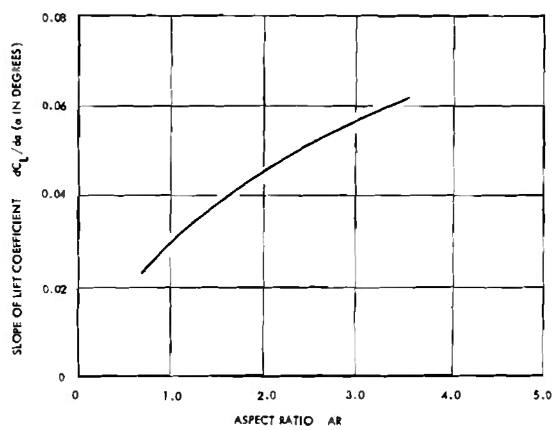


Figure 7-101. Slope of Lift Coefficient as Function of Aspect Ratio for Rectangular Wings

stall angles which are associated with these maximum lift coefficients are shown in Fig. 7-103 from the same reference. Note that the stall angles for low aspect ratio foils are higher than for high aspect ratio foils. High aspect ratio foils have a higher lift coefficient slope, but their lower stall angles usually result in lower maximum lift coefficients. The thickness-chord ratio has an effect on the maximum lift coefficient of a foil, as discussed in Ref. 67. The thickness of a balanced rudder is seldom less than 15 percent of the chord.

In practice, it is very difficult to determine the stall angle and maximum lift coefficients for rudders on a ship. The flow past the rudder is turbulent so that larger stall angles are found than for foils in undisturbed flow. Interaction with the hull increases the effective aspect ratio of the rudder. Free surface effects are also present. Model tests for stalling are of little use because of serious scale effects. Scale effects may be considered negligible for rudder angles below the stall angle, however.

Lift coefficient data from model tests on spade type rudders are presented in Ref. 106. An empirical fit to these data gives the following equation for the lift coefficient slope C_{L_a} with respect to angle of attack for small angles:

$$C_{L_a}\Big|_{a=0} = \frac{dC_L}{da} = \left[\frac{0.0988\,A_e}{1.8 + \cos\Omega\sqrt{\dfrac{A_e^2}{\cos\Omega} + 4}}\right] \qquad (7\text{-}140)$$

where

A_e = effective aspect ratio and is just twice the value in Eq. 7-139

Ω = sweep angle of the one-quarter chord line, deg

a = rudder angle of attack, deg

The flow velocity past the rudder must include the effects of both the hull wake and the propeller slipstream. If we assume that the ultimate contraction of the slipstream occurs ahead of the rudder, then the mean velocity and diameter of the slipstream are given (by momentum theory):

$$v_1 = \sqrt{V_a^2 + \frac{2T}{\rho A}} \qquad (7\text{-}141)$$

$$D_1 = D\sqrt{\frac{v_1 + V_a}{2v_1}} \qquad (7\text{-}142)$$

where

D = propeller diameter, ft

D_1 = slipstream diameter, ft

v_1 = mean slipstream velocity ft/sec

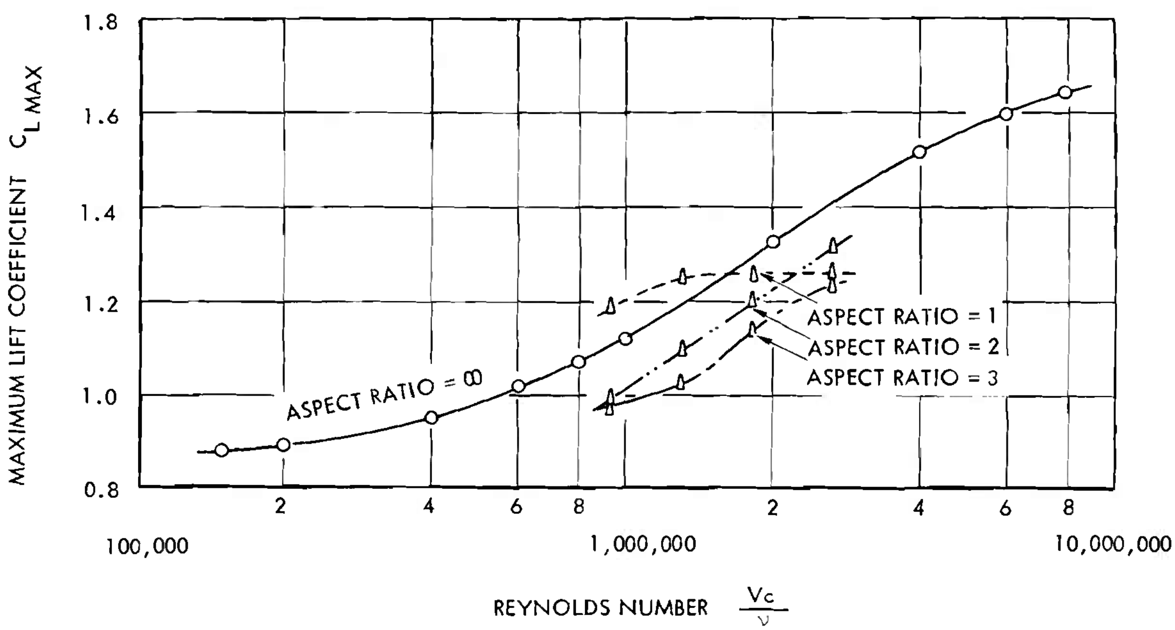


Figure 7-102. Effect of Reynolds Number on Maximum Lift Coefficient for Various Aspect Ratio Foils

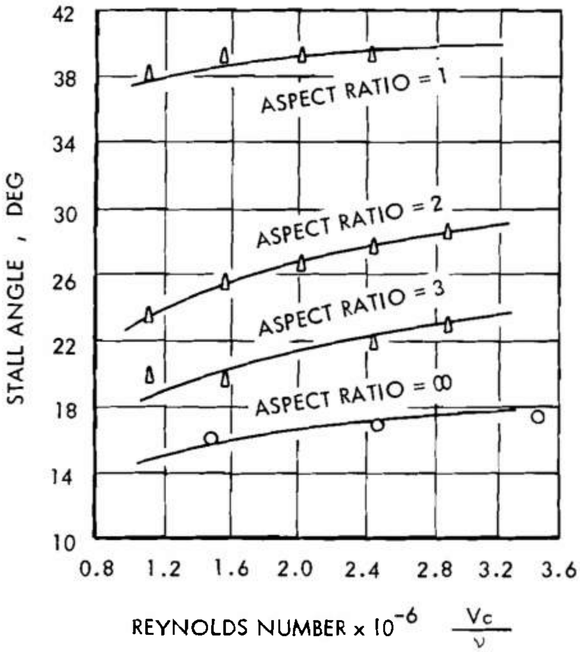


Figure 7-103. Effect of Reynolds Number on Stall Angle for Various Aspects Ratio Foils

V_a = advance speed of the propeller, ft/sec
T = propeller thrust, lb
A = propeller disc area, ft^2
ρ = mass density, slug/ft^3

With these values, the following equation may be used to calculate the mean velocity over the rudder surface:

$$v = \frac{D_1 v_1 + V_a (D_1 - b)}{b}, \text{ ft/sec} \qquad (7\text{-}143)$$

where

b = span of the rudder, ft.

If the rudder is behind a Kort nozzle, of course, the assumption about the slipstream contraction will not be valid. In this case the slipstream diameter should be taken as the propeller diameter and the mean slipstream velocity as given by Eq. 7-141, where T is propeller thrust and does not include the thrust contributed by the nozzle.

7-9.2.2.2 Geometrical Considerations

Geometric parameters which must be considered in the design of a rudder include:

a. *Aspect Ratio.* The magnitude of the aspect ratio is limited by considerations of vulnerability. The rudder should not extend below the base line unless there is some other projection such as a skeg or Kort nozzle which affords adequate protection from likely hazards. Structural limitations must also be considered in selecting aspect ratio. The center of pressure is generally about midway between the root and tip of the rudder. Thus, the greater the span, the greater will be the bending moment on the rudder stock.

From the hydrodynamic standpoint, increasing the aspect ratio increases the slope of the lift curve with respect to angle of attack. Thus, the rudder will be more effective at low rudder angles so that less rudder angle is required to counteract deviations in heading (with the same rudder area).

The root of the rudder should be as close to the hull as possible. If the gap between hull and rudder is sufficiently small, the effective aspect ratio is theoretically twice its geometric aspect ratio.

b. *Planform and Section Shapes.* Two principles govern the selection of the rudder planform. First, the advantages of elliptical spanwise loading are minimum induced drag, reduced bending moments, lower critical cavitation numbers, and increased lift curve slope. Elliptical loadings can be achieved by selecting the sweep angle of the quarter chord line with respect to the taper ratio as given in Ref. 105. Of more practical importance is the second principle, namely, to keep the largest possible proportion of the rudder area in the propeller slipstream.

A suitable section for all-movable rudders is the NACA symmetrical section. The ordinates for this section are given in Table 7-46.

The designation NACA 0015 refers to a NACA symmetrical section with a thickness to chord ratio t/c = 0.15.

The spanwise thickness distribution usually tapers from a maximum at the root ($t/c \approx 0.18$) to a minimum at the tip ($t/c \approx 0.04$).

c. *Balance.* The objective in balancing the rudder is to minimize the rudder torque. Low torque is, of course, desirable to minimize the size and weight of the steering gear. It is not possible to obtain full balance (zero torque) at all rudder angles because the center of pressure moves aft with increasing angle of attack. Thus,

TABLE 7-46 OFFSETS FOR NACA SYMMETRICAL SECTIONS

Chord from Leading Edge, %	Percent of Maximum Section Thickness
0	0
2.5	43.6
5	59.3
7.5	70.0
10	78.1
15	89.1
20	95.6
25	99.0
30	100.0
40	96.7
50	88.2
60	76.1
70	61.1
80	43.7
90	24.1
100	2.1

if a rudder were designed to have zero torque at 20 deg rudder angle, the torque would be negative at lower rudder angles and positive at higher angles. Ideally, the maximum negative torque should equal the maximum positive torque in order to minimize the torque. A positive torque is defined as one which opposes an increase in rudder angle. A hand-steered vessel cannot have a negative rudder torque at small angles or the rudder will not trail at zero angle when the rudder is free. With power steering, negative rudder torques are accepted in order to lower the maximum positive torque.

The balance of the rudder is expressed in terms of the percentage of the total rudder area which is forward of the rudder turning axis. This percentage will range between 20 and 30 percent to achieve equal positive and negative maximum torques. The optimum area balance depends on the disposition of rudder area, section shapes and thickness, the effect of propeller race on the rudder center of pressure, and the maximum rudder angle.

The maximum rudder torque in the astern condition depends on the maximum rudder angle and speed, and the degree of rudder area balance. It may be desirable to limit the allowable rudder travel in the astern condition so that the torque does not exceed the maximum ahead torque. Otherwise, the astern torque can be reduced by increasing the balance area so that the rudder is overbalanced in the ahead condition.

d. *Stock Angle.* It was found that the effectiveness of the rudder on the DUKW was increased considerably by raking the rudder axis aft. This rudder was in a tunnel so that there was a large interaction between the rudder and hull. By raking the rudder, the flow was deflected down and clear of the tunnel at large rudder angles to reduce the interaction.

e. *Maximum Rudder Angle.* The effect of aspect ratio on the maximum lift force that a rudder can develop has not yet been determined. It can be shown, however, that rudder angles greater than the stall angles in Fig. 7-103 can be used effectively to decrease an amphibian's turning diameter. The explanation for this lies in the fact that the angle of attack α (Fig. 7-97) is less than the rudder angle δ in a turn. If rudder angles greater than the stall angle can be attained, the helmsman must be careful to avoid inadvertent stalling of the rudder. If the rudder is put hard over so that it stalls, there will be a large increase in rudder drag without the desired maximum lift for turning. For this reason, it is advisable to have a simple rudder angle indicator on the control panel for the operator's use. Maximum rudder angles as high as 65 deg may be practical, but should only be considered if a small turning diameter is essential.

f. *Hull Geometry.* The shape of the hull in the vicinity of the rudder can have an important effect on rudder performance especially if the propeller and rudder are placed in a tunnel. This topic is discussed in par. 7-1.4.

7-9.2.3 Trainable Propulsors

The side force that a trainable propulsor can exert at zero advance speed is just equal to its static thrust. With a steering angle of 90 deg, the turning moment is simply the thrust multiplied by the moment arm from the propulsor to the amphibian's center of gravity.

7-9.2.3.1 Water Jets

The side force of a trainable water jet at some advance speed can be estimated from simple

momentum theory. The thrust coefficient C_T (neglecting interactions with the hull) may be written

$$C_T = \frac{T}{\frac{1}{2}(\rho A V_a^2)} = 2V_j'(V_j' - 1) \qquad (7\text{-}144)$$

where
- T = water jet thrust, lb
- A = jet area, ft^2
- V_a = advance speed at the water scoop inlet, ft/sec
- V_j = jet velocity, ft/sec
- V_j' = ratio of jet velocity to advance speed, nd
- ρ = mass density, slug/ft^3

Similarly, a lift coefficient corresponding to the side force may be written

$$C_L = \frac{L}{\frac{1}{2}(\rho A V_a^2)} = 2(V_j')^2 \sin a \qquad (145)$$

where
- L = component of thrust normal to the vehicle centerline, lb
- a = "rudder" angle of jet, deg

Eqs. 7-144 and 7-145 can be combined to give the slope of the lift coefficient curve with respect to angle (in degrees) as:

$$C_{L_a}\Big|_{a=0} = \frac{C_T}{57.3}\left[\frac{\sqrt{1+2C_T}+1}{\sqrt{1+2C_T}-1}\right] \qquad (7\text{-}146)$$

7-9.2.3.2 Outdrive Units

The lift force of a right angle drive with an open propeller will not be considered here because it is believed more likely that a right angle drive unit designed for a future amphibian will have a Kort nozzle propulsor. Annular foils have been tested in a wind tunnel to investigate their lift characteristics, Ref. 107. The results of these experiments indicate that the lift coefficient of a shroud in free stream flow is about twice the lift coefficient of a conventional low aspect ratio rudder (Eq. 7-140) where the aspect ratio AR and projected areas S of the shroud are defined, respectively, as

$$AR = d/c, \text{ nd}$$
$$S = dc, \text{ ft}^2 \qquad (7\text{-}147)$$

where
- d = inside diameter of the shroud, ft
- c = length of the shroud, parallel to the centerline, ft

Eq. 7-140 may be simplified (Ω = 0) and increased by a factor of two to yield an equation for the slope of lift coefficient with respect to angle of attack for an annular foil in free stream flow:

$$C_{L_{o_a}}\Big|_{a=0} = \frac{dC_{L_o}}{da}\Big|_{a=0} = \frac{0.1976\,AR}{1.8+\sqrt{(AR)^2+4}} \qquad (7\text{-}148)$$

where
- $C_{L_o} = \dfrac{L}{\frac{1}{2}\rho S V_a^2}$ = lift coefficient for a shroud in free stream flow, nd
- L = lift or force normal to direction of flow, lb
- S = projected area of shroud, ft^2
- V_a = advance speed, ft/sec
- ρ = mass density, slug/ft^3

This equation gives a good fit to the experimental data in Ref. 107.

The lift of a Kort nozzle shroud will be influenced by the flow velocities induced by the propeller. A rough estimate of this effect can be made from simple momentum theory by assuming that the effective velocity past the shroud is the advance speed augmented by one-half the velocity increment in the slipstream. Thus,

$$C_{L_a} = C_{L_{o_a}}\left(\frac{1+\sqrt{1+C_{T_p}}}{2}\right)^2 \qquad (7\text{-}149)$$

where
- C_{L_a} = slope of the lift coefficient curve for a shroud enclosing a thrusting propeller, nd
- $C_{L_{o_a}}$ = slope of the lift coefficient curve for a shroud in free stream flow, nd
- C_{T_p} = thrust coefficient of the propeller, nd

$$C_{T_p} = \frac{T_p}{\frac{1}{2}(\rho A V_a^2)} \qquad (7\text{-}150)$$

where

T_p = portion of total thrust produced by propeller, lb

A = $\pi D^2/4$ = propeller disc area, ft^2

D = propeller diameter, ft

V_a = advance speed, ft/sec

ρ = mass density, slug/ft^3

To complete the picture, one must consider the propeller thrust component normal to the ship

$$C_{L_T} = \frac{T_p \sin \alpha}{\frac{1}{2}(\rho S V_a^2)} = C_{T_p}\left(\frac{\pi D^2}{4}\right)\left(\frac{1}{cd}\right)\sin \alpha$$

$$\text{If } D = d,\ C_{L_T} = C_{T_p}\left(\frac{\pi}{4}\right)\left(\frac{d}{c}\right)\sin \alpha \qquad (7\text{-}151)$$

$$\text{Then, } C_{L_{\alpha_T}} = 0.0137\,(AR)\,C_{T_p} \qquad (7\text{-}152)$$

The total lift coefficient slope may be estimated by adding Eqs. 7-149 and 7-151, i.e., $C_{L_\alpha} + C_{L_{\alpha_T}}$

Refs. 108 and 109 present model test data for the normal force of Kort nozzle configurations. One study was made for the design of positioning units for the Project MOHOLE Drilling Platform and the other was made for the design of a tilt-duct VTOL craft.

7-9.2.4 Ducted Thrusters

The static thrust produced by a transverse tube type thruster may be estimated from

$$T = [C^2 P^2 \rho A]^{1/3}, \text{ lb} \qquad (7\text{-}153)$$

where

P = delivered power, ft-lb/sec

A = disc area, ft^2

C = figure of merit, nd

ρ = mass density, slug/ft^3

Ref. 110 shows that the value of C for tunnel thrusters is approximately 0.8. This may be compared with the value of C for Kort nozzles which may be as high as 1.4. Thus, for the same power and diameter, the static thrust produced by a Kort nozzle will be about 40 percent greater than by a tunnel thruster.

The side force of a tunnel thruster is seriously degraded when the ship is underway. The effect of forward speed on the thrust of such a device depends greatly upon flow across the tunnel openings, which is a function of the hull shape. There are no data available for the performance of a tunnel thruster installed in a hull similar to an amphibian. However, based on available data, it is not unreasonable to expect that the side force produced by a tunnel thruster in the bow of an amphibian at cruise speed would be less than 30 percent of its static thrust.

7-9.2.5 Box Rudders

In order to improve the docking performance of the LARC V amphibian, model tests were conducted at the Davidson Laboratory of the Stevens Institute of Technology, Ref. 111. Several rudder configurations were tested:

a. Existing single flat plate rudder
b. Three-parallel-bladed rudder
c. Three-tandem-bladed rudder
d. Circular nozzle* with airfoil section
e. Circular nozzle* with flat plate sections
f. Elliptical plate nozzle*
g. Rectangular plate nozzle*
h. Rectangular plate nozzle* with center plate

*These are more accurately described as tube rudders; they are placed behind the propeller and do not function as a Kort nozzle. The propeller was enclosed in a fixed nozzle, independent of the rudder configuration.

Measurements of side force and yawing moment were made in the static condition; the propeller produced thrust, but the model was constrained so that there was no motion. The various configurations were compared on the basis of a steering performance index which was simply the rudder's lift curve slope (side force per degree of rudder angle). On this basis, the rectangular plate nozzle with center plate (box rudder) was the most effective, yielding some 72 percent improvement over the existing rudder configuration.

The large improvement in side force (per degree of rudder deflection) realized by the box rudder configuration is due largely to the simple fact that more rudder area is placed in the propeller slipstream. This rudder configuration was adopted for the LARC XV and is now standard equipment.

7-9.2.6 Steering Linkages

It is important that the waterborne steering

system be coordinated with the overland steering system. This coordination allows the transition between waterborne and overland operations to be made smoothly and safely. In addition, as was pointed out earlier, it is desirable to use the wheels as rudders for operations through the surf where steering may alternately be accomplished either by rudder or wheels, or both. These considerations dictate that water and land steering systems be interlocked, and that steering be accomplished by a single control. The linkage should be designed so that the maximum deflection of the rudder will correspond with the maximum cramp angle of the wheels.

The large forces necessary to turn a 40,000-lb amphibian on land will dictate that power steering be employed. Hence, both land and waterborne steering will likely be hydraulically powered.

The strength of the waterborne steering system linkage should be based on the maximum rudder torque, whether it be from ahead or astern operation. The maximum power P which must be supplied is a product of the maximum ahead rudder torque and the desired rudder angular velocity:

$$P = \frac{Q\omega_r}{550(57.3)}\text{, hp} \qquad (7\text{-}154)$$

where

Q = torque, ft-lb

ω_r = rudder angle rate, deg/sec

The optimum angular velocity of the rudder depends, to a large extent, on the quickness of the ship's response to the rudder. A fast rudder deflection rate is of little use if the ship responds so slowly that the rudder stall angle is almost immediately exceeded. If the ship's response is quick so that the drift angle ϕ (Fig. 7-97) increases rapidly with an increase in rudder angle δ, then the angle of attack α will not exceed the stall angle. Ref. 105 suggests that a suitable rudder rate may be

$$\omega_r = \frac{60\ V}{L} \qquad (7\text{-}155)$$

where

ω_r = rudder angle rate, deg/sec

V = ship speed, ft/sec

L = ship length, ft

If we apply this formula, a 40-ft amphibian with 8 mph speed should have a rudder rate on the order of 17 deg per sec.

The rotational velocities in the propeller slipstream can cause the rudder to have a lift force when it is placed amidships (zero rudder angle). This effect was noticeable on the DUKW. As a result, the steering linkage on the DUKW was modified so that when the helm angle was zero the rudder was at the angle which gave zero turning moment. The linkage was also arranged so that the ratio of rudder travel to helm angle was greater on one side of the zero helm angle than the other. This was done so that the helm would have equal effectiveness to port and starboard even though the rudder did not.

7-9.3 WATERBORNE STEERING BY MEANS OF THE WHEELS

It has already been noted that the wheels can be used effectively as rudders in the surf mode of operation. Their effectiveness is improved by placing faired covers over the hubs which also decreases their form drag. Close attention to the hull design in way of the wheel wells can also improve the flow past the wheels so that their effectiveness as rudders is increased.

Many amphibians, and especially high speed designs, have provisions for retracting the wheels and/or extending flaps or fairings to "clean up" the flow around the wheel wells. These provisions are all directed toward decreasing the form drag which is a large part of the total drag on the amphibian. When the amphibian is operating in the hull mode or some high speed mode with wheels retracted or covered with flaps, the wheels cannot be used as rudders and some provision should be made to automatically disconnect the overland steering system from the waterborne steering system when the wheels are retracted.

7-9.4 OVERLAND STEERING SYSTEM

The overland steering system, in conjunction with the suspension system, determines the degree of control and stability for the amphibian on land. The turning radius is a function of the amphibian geometry and the type of steering employed. Figs. 7-104 and 7-105 show the geometry of various turning radii for conventional Ackermann (2-wheel) steering used on all standard military vehicles and for 4-wheel

steering, respectively. The angle β is the maximum cramp angle for the inside wheel in a minimum radius turn. The angle α is the cramp angle for the outside wheel. The relationship between α and β is chosen so that the axis of each wheel goes through the center of curvature of the vehicle path. Otherwise, there would be excessive lateral scrubbing of the tires in a turn. The relation between the front wheel steering angles for Ackermann steering is:

$$\cot\beta = \cot\alpha - \frac{W}{L} \tag{7-156}$$

where

L = wheelbase, ft

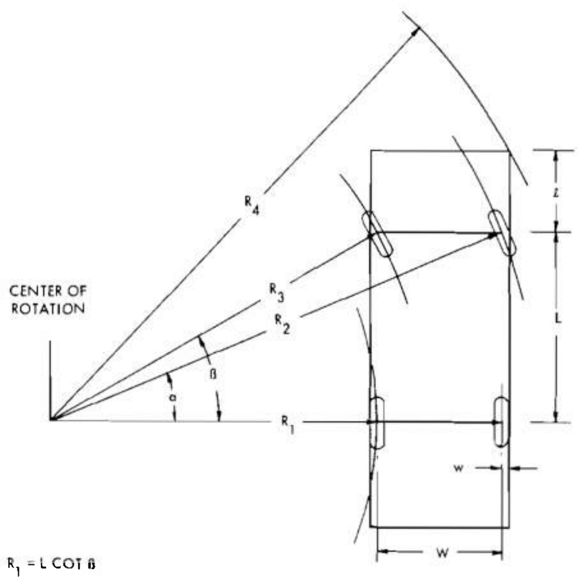
W = length between steering pivots, ft

It can readily be shown that the maximum cramp angle of the front wheels to effect a given radius turn is smaller with four-wheel steering. The maximum cramp angle is of importance for an amphibian design because it determines the depth of the wheel well cutouts. Deep wheel wells are undesirable with respect to drag, buoyancy, arrangements and, especially, strength of the hull. Amphibians with four-wheel steering can be made even more maneuverable by making provision for crab steering, as shown in Fig. 7-106. Crab steering is indispensable for maneuvering a large cumbersome vehicle in close quarters. Of course, the advantages of four-wheel and crab steering are offset to some extent by the increased complexity of the system.

Other factors to be considered in the design of wheeled amphibian steering systems are effects of braking and traction, suspension geometry, inertial and aerodynamic forces, powering, gearing, and alignment. These factors are all discussed in Ref. 63.

7-9.5 LOCATION OF THE STEERING POSITION, CONTROLS, AND INDICATORS

The operator's location on an amphibian will be a compromise between the optimum position for steering on land and steering on water. On land the most desirable position is as far forward as possible so that the operator can see over the bow in order to avoid obstructions. On water, however, safe operation dictates that the operator's station be placed farther aft to afford protection from the impact of the surf and also to improve the helmsman's "feel" of the rudder.



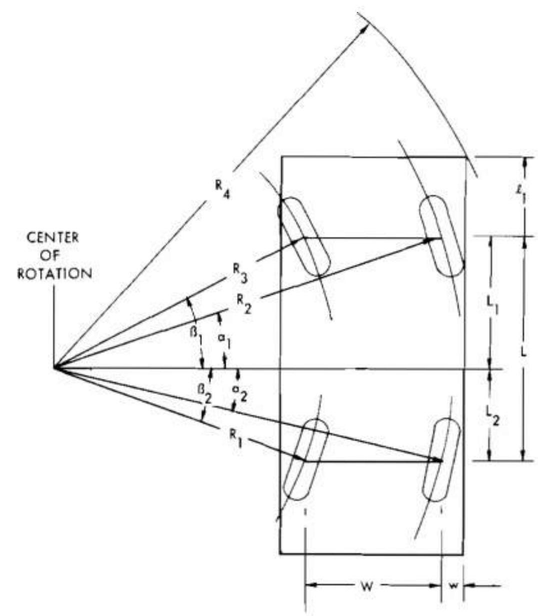
$R_1 = L \cot\beta$

$R_2 = \left[(R_1 + w)^2 + L^2\right]^{1/2}$

$R_3 = L/\sin\beta$

$R_4 = \left[(R_1 + W + w)^2 + (L + \ell)^2\right]^{1/2}$

Figure 7-104. Ackermann Steering



$L_1 = L_2(\tan\beta_1 / \tan\beta_2)$

$R_1 = L_2 / \sin\beta_2$

$R_2 = \left[(L_1 \cot\beta_1 + w)^2 + L_1{}^2\right]^{1/2}$

$R_3 = L_1 / \sin\beta_1$

$R_4 = \left[(L_1 \cot\beta_1 + W + w)^2 + (L_1 + \ell_1)^2\right]^{1/2}$

Figure 7-105. 4-wheel Steering

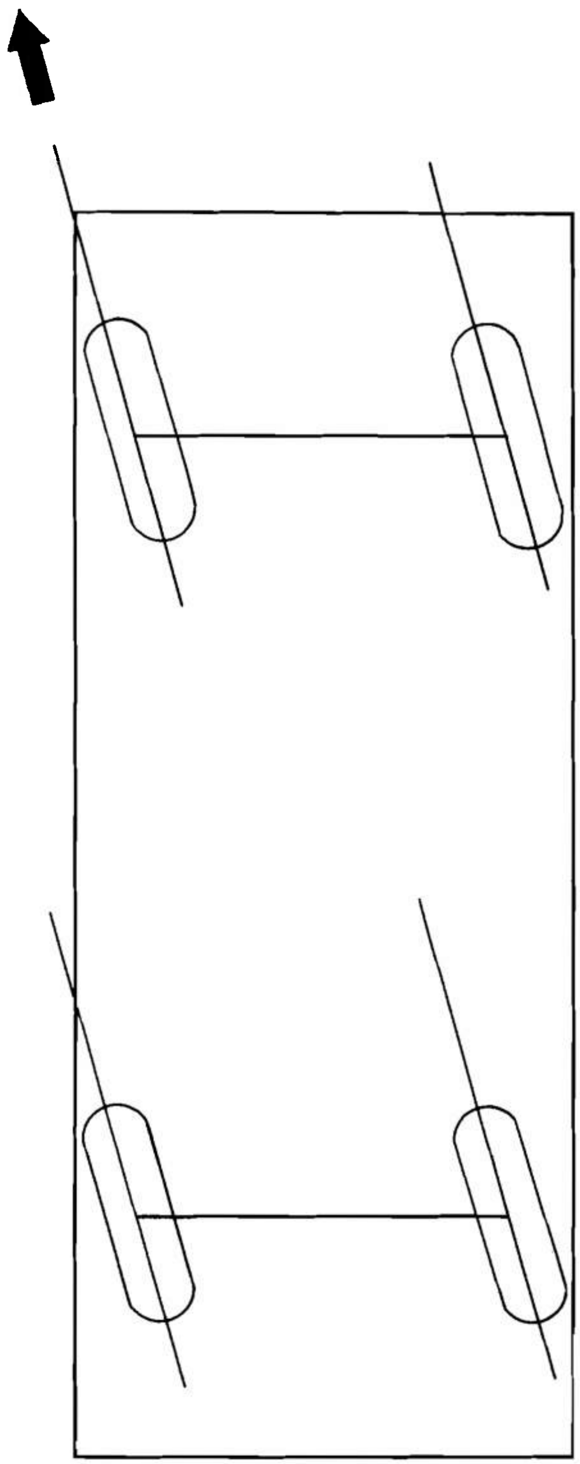


Figure 7-106. Crab Steering

On a ship, the operator senses changes in heading by sighting across some reference point on the bow at a point on the horizon. The spatial relationship between these two points indicates to the operator any change in heading and also the rate of the change. The sensitivity of this type of navigation increases as the distance between the operator and his visual reference point on the ship increases. It should be noted that the inner ear is virtually useless in sensing changes in heading on a tossing vessel. This, along with the fact that there is no feel to the rudder with power steering, points up the need for a rudder angle indicator.

The LARC XV was designed to head in one direction at sea and the other direction on land in an attempt to solve the conflicting requirements for operator location. In this manner, the cab was positioned on the extreme forward end for good road visibility on land and aft for protection at sea. This was not a completely satisfactory solution to the problem, however. Upon reaching land through the surf, the amphibian must come to a stop while the operator changes seats (the seats face in opposite directions, and one of them is elevated to place the operator in a bubble on top of the cab for complete 360 deg visibility at sea) and turns the amphibian around for land operation. The sequence of operations is reversed for entry into the water. This arrangement led to increased complexity in the controls which had to be used for both modes of operation, and certain instruments and indicators were difficult to monitor from one seat or the other.

The LARC V has only one steering position which is placed farther aft than would be considered as ideal for overland operations. As a result, an assistant must climb out of the amphibian to direct the operator over difficult terrain.

An alternative design which should be studied for future amphibians would place the cab far forward. An auxiliary steering station would be located on the aft end of the amphibian. This station would have duplicate controls and instruments needed for safe waterborne operation, and would only be manned during operations in heavy surf and high seas. The feasibility of this type of solution would require further study.

The location of the controls should be

convenient to the operator, and all instruments and indicators should be in full view and easily read and interpreted. The operator's seat should be adjustable to suit his particular anatomy. The best way to insure against a poor design in which certain vital controls are out of reach and instruments cannot be seen is to build a full scale mock-up of the control station which can be evaluated by experienced amphibian operators.

SECTION V AUXILIARY SYSTEMS

7-10 AUXILIARY SYSTEMS DESIGN

Amphibians are generally multi-mission vehicles operated by a crew of from two to five men. The auxiliary systems must therefore provide basic, remote control, and labor saving services so that the crew-vehicle combination can provide a flexible, cost-effective system. The auxiliary systems necessary to accomplish this goal are numerous, and in some instances, complex. This section of the handbook does not explain the design procedure for each system but it does consider the problems generated by the marine environment and reviews, for each system, the important lessons learned from recent amphibians.

7-10.1 HYDRAULIC SYSTEMS

Hydraulic systems are used to transmit power and to provide remote controls. Amphibian hydraulic systems could transmit and provide power for:

a. Steering
b. Braking
c. Wheel extension and retraction
d. Ramp extension and retraction
e. Winch drives
f. Miscellaneous drives

Hydraulic remote control is normally used only in conjunction with a power transmitting system. Components of a system include:

a. Source of mechanical power such as an electric motor, power take-off (PTO) from the main engine, or manual

b. Power converter (mechanical to hydraulic) such as a rotary or reciprocating pump

c. Hydraulic power transmission system which includes valves, tubes or pipes, regulators, and manifolds

d. Power converter (hydraulic to mechanical) such as a hydraulic cylinder, rotary actuator or hydraulic motor

e. Miscellaneous components such as reservoirs, filters, heat exchangers.

Combinations of these components make up the hydraulic system. Frequently a single hydraulic pump is used to power a number of components.

The design techniques and use of fluid-power and fluid-control systems are expanding rapidly. This implies that present design criteria will change as new components are produced and a larger production base evolves. *Machine Design* magazine publishes annually a reference issue devoted entirely to fluid power which contains current information on hydraulic system design. The Mobility Equipment Research and Development Center, Fort Belvoir, Virginia, maintains the Qualified Products List (QPL) of hydraulic equipment. The QPL's should not be used without consulting the maintaining authority. Some of the design guidelines in use at present (1969) are described in the paragraphs which follow.

For a required performance level, the system weight tends to decrease with an increase in system pressure. The cost and maintenance, however, tend to increase with system pressure. Most system pressures do not exceed 3,000 psi and the variety of available components for higher pressures is limited. Component availability is greatest at 2,000 to 2,500 psi.

Systems in which the pump runs continuously may be either open or closed center systems. When the drive end of the systems is not using fluid, the open center configuration discharges the fluid back to the reservoir through the control valve, while the closed center system returns the fluid through the pressure relief valve. This means, of course, that when there is no power required and the system is on "stand-by", the open-center configuration operates at a reduced pressure and pump power level while the closed-center system pump maintains a full head of pressure. The

closed-center configuration is used where there are two or more branches in the circuit, operating independently. It is possible to use an open-centered configuration to power more than one cylinder or motor independently if a series arrangement is used. Special precautions are required because the tankport on control valves is usually not designed for full system pressure. In general, the closed system is preferred for multiple branch systems where at least one of the motors is used continuously. Open-center configurations are preferred for simple systems which are not used continuously.

Another important system consideration is the selection of the hydraulic fluid. Ref. 112 is a handbook on hydraulic fluids.

Simple low pressure hydraulic systems are easy to understand, operate, and maintain. Systems on amphibians, however, are complex and are frequently hampered with fluid dynamic problems such as cavitation. In order to obtain a cost-effective trouble-free system, it must be completely designed and tested by engineers who have experience with similar systems in a similar environment. In the past, attempts to reduce the cost using components with marginal capabilities have resulted in systems with low reliability and the added expense of additional development time.

The source of mechanical power is frequently a mechanical power take-off (PTO) from the main engine which is normally a standard optional feature. The power converter is usually a simple piston pump for manual operation or one of a variety of types of motor-driven pumps. Ref. 113 contains a list of available types of motor-driven pumps and a discussion of the general characteristics of each type. Gear pumps were used on the LARC V and LARC XV. In spite of serious development efforts, these pumps had problems which resulted in unpredictable operation and short lives. It is recommended that these problems be reviewed before using this type of pump.

Hydraulic power is transmitted primarily through rigid tubing. Steel is the only tubing material permitted without restriction by the "Joint Industry Conference, Hydraulic and Pneumatic Standards for Industrial Equipment". Tube fittings are selected for minimum resistance to flow in the system, reliability in the environment, resistance to leakage, and life cycle cost, including the initial cost of parts and labor and the maintenance cost. Fittings are made of steel and are connected with mechanical seals which may be flanged, flared, or flareless types. Flanged seals are used only in the larger sizes. Flared tubing for hydraulic applications is normally used with the JIC-SAE 37 deg flare. A 45 deg SAE flare is frequently used for low pressures with copper tubing. For steel tubing, the flareless fitting is leakproof to very high pressures and resists vibrations well. If flared type fittings are used, the nut-and-sleeve type is preferred. Thin wall stainless steel tubing work hardens readily and both the tube-cutter and the various flaring tools can work-harden the tube so that either flareless or flared fittings leak at moderate pressures.

The hull of any vehicle flexes and some machinery components will rock on their foundations so that it is important to arrange the tubing to allow for motion between components. If the relative motions are large, flexible hose should be used. In addition to permitting relative motion, hose can accommodate the following environmental conditions:

a. Severe vibrations
b. Manufacturing tolerances
c. Complex routing for piping runs
d. Hydraulic shock absorption

Flexible hose changes length when pressurized so that it should not be installed in tension. It fails more quickly from twisting than bending so it is desirable to support the hose in such a manner that the motion in any one section is in one plane.

Valves can be classified as follows:

a. Pressure control
b. Flow control
c. Direction control

The flexibility of hydraulic systems is in large measure the result of the variety of control valves available. This constitutes a large complex subject and the manufacturers should be consulted as to the characteristics of available valves. Automatic pressure regulating valves have been a high maintenance component on some past amphibians. In general, complicated control valves cannot be expected to be highly reliable in the amphibian environment. For Navy hydrofoil boats, packaged hydraulic systems have performed better than those assembled at the boat yard.

Hydraulic power is converted back to

mechanical force by:

a. Linear actuators

b. Hydraulic motors

c. Rotary actuators

Linear actuators include all of the types of hydraulic cylinders. These have high force capability, high power per weight and size, mounting flexibility, and ease of speed control.

Hydraulic motors include a variety of variable and constant volume configurations. Motor-type comparisons are presented in Ref. 113.

Rotary actuators are either vane or piston types. Typical maximum arcs of rotation are 280 deg for single vane and 100 deg for double vane types. Piston-type rotary actuators can exceed 360 deg rotation.

The miscellaneous accessories for a hydraulic system are sometimes neglected since they generally serve only an auxiliary function. Reservoirs, one of the more important of these items, are sized to hold the full flow from the pump for from one to three minutes. Factors which affect the size are:

a. Volume to fill the system at start-up without exposing the intake filter

b. Capacity to accommodate variations in the system requirements during each cycle of the operation

c. Heating and cooling dynamics during extreme temperatures

Other factors that must be considered in selecting the reservoir are:

a. Line location at extreme angles of heel and trim

b. Baffles to reduce surging in the tank

c. Provisions for draining and cleaning the tank

d. Level indicator

e. Air filters to remove the contaminants from the air as the tank breathes

Reliable hydraulic systems require clean hydraulic fluid. Filters are required, even if the system is initially clean, because of the many sources of contamination. Filters are available in a variety of types. The selection criteria include:

a. Fluid characteristics

b. Solids to be removed

c. Piping material

d. Pressure

e. Environment

All hydraulic fluids have limiting values of practical operating temperatures. For systems which operate continuously in normal environmental conditions, a heat exchanger will be necessary to cool the hydraulic fluid. These heat exchangers usually use the same fluid that cools the main engines. Finned tubing has been used effectively for this purpose. In general there is very little air flowing over the reservoir so that the heat exchanger should be designed to carry off the full amount of heat generated in the system.

7-10.2 ELECTRICAL SYSTEM

Electricity is required for lights, horn, controls, gages, electronic equipment, and power for small motors. Components in the electrical system of a typical amphibian include:

a. Batteries

b. Starter solenoid

c. Alternator

d. Starter motor

e. Engine fan disconnect switch

f. Fuel shut-off valve

g. Junction box with contactors, circuit breakers, relays, and a shunt

h. Fuel pump motor

i. Lights, including headlights, taillights, running lights, bow light, stern light, cab lights, machinery space lights, instrument lights

j. Instruments

k. Windshield wiper

l. Radio

m. Horn

n. Slave receptacle

o. Heater fan motor

p. Ramp horn

q. Emergency hydraulic pump motor

The major differences between electrical systems for amphibians and those provided for trucks are that stray current in a metal amphibian can accelerate electrolysis, and salt water spray and salt-laden air accelerate the corrosion of contacts and the deterioration of insulation. Electrolysis caused by stray currents is practically eliminated by using a two wire system where the metal hull is not used as one side of the circuit. Two wire systems, however, are heavier and more expensive. In order to prevent premature failure of the system, it is necessary to select satisfactory materials for the environment and encapsulate or otherwise waterproof the entire system.

Most boats are fitted with a 6- or 12-V system but marine equipment is frequently available for

24-V service. At the present time 24-V DC systems are standard for all military vehicles. This voltage is prescribed by the Department of the Army in AR 705-19, Ref. 114. Refs. 115 to 119 contain information pertaining to the design and requirements of electrical systems for boats.

Ref. 63 contains discussions of electrical systems for military vehicles. The U. S. Army Tank-Automotive Command, Warren, Michigan, maintains the list of qualified electrical products for use in Army vehicles.

Some design problems and/or solutions from recent amphibian development program are:

a. Wires and components must not be located in the bilge, in areas where there may be excessive spray, or where they will be susceptible to damage by maintenance personnel.

b. Alternators are simpler than marine (nonsparking) DC generators.

c. Most prevalent failures have been waterlogged switches and contacts, and insulation deterioration.

d. In cold weather the batteries may require heating.

e. A receptacle and extension cord for jumping electrical systems between vehicles are desirable.

Some of the present amphibian operators find a high-intensity searchlight useful for finding the low spots in the surf when coming from the sea into shore. These should be located on the cab roof and could be either fixed in position or manually trainable.

7-10.3 FUEL SYSTEMS

Fuel systems for gasoline, or other fuels with a flashpoint of 110°F or lower, must be designed with more safeguards than required when using Diesel fuel. Gasoline vapor is heavier than air and tends to collect at the low point of the bilge. In a truck installation the vapor flows out the bottom and is dissipated. If the concentration is within certain bounds, any spark or open flame will cause an explosion. A safe fuel system should be:

a. Liquid and vapor tight to the interior of the hull

b. Permanently installed with all components properly secured

c. Accessible for inspection and maintenance, particularly the valves and filters or strainers

d. Adequately protected from high pressures caused by thermal expansion of the fuel when the system is secured

e. Properly bonded across components that will not carry electricity and grounded so that all metallic components have a low resistance continuity to ground, thus preventing static electricity sparks

f. In areas that are not subject to accumulation of water which might eventually leak into the system

g. Able to withstand exposure to free burning fuel for a specified time so that the crew can escape in event of fire (Ref. 115 recommends 2.5 min for yachts.)

h. Constructed of materials that are compatible with the fuel

i. Built to withstand the pressures, vibration, shock, and other environmental conditions

j. Fitted with the necessary filters and fuel pick-up tubes since the vehicle motions may be extreme

k. Fitted with connections only on the top to reduce the amount of fuel leakage in case of a fitting failure

l. Fitted with a fill pipe or pipes that open only on deck, with a watertight cap, in a location where spilled fuel and/or fumes will flow overboard

m. Fitted with vent lines that have flame screens to prevent flashbacks

n. Arranged such that a broken distribution line cannot siphon the fuel from the fuel tank into the bilge

o. Able to prime engines for cold starts or after running out of fuel

These vehicles will occasionally be fueled alongside a ship so the location of the fill pipes must be safe for personnel and free from water washing into the fuel system. Fill pipes must be sized so that they do not restrict the refueling operation. Vents must be located so that the fuel vapor flows overboard and, in heavy weather, sea water does not flow back through the vent line. Vent lines should not be tapped into fill lines.

Rigid fuel tanks may be constructed from steel, aluminum, or Fiberglas. Each has advantages and disadvantages which must be evaluated with respect to the particular design. Flexible bladder fuel tanks have also been used to permit fitting into locations which are

inaccessible to rigid tanks. However, they are susceptible to fire and abrasion.

Electrically driven submerged fuel pumps are dependable and tend to avoid vapor lock.

7-10.4 COOLING SYSTEMS

Boat machinery cooling system heat exchangers are located under or inside the hull. If under the hull, it is normally located alongside the keel where it is well protected and is usually defined as a "keel cooler". For installations with the heat exchanger inboard, the seawater is pumped directly through the engine and the other components or through a heat exchanger. For craft which operate in the surf, such as landing craft or amphibians, where there is a large quantity of suspended solids in the water, a system that pumps water through a piping system is impractical. The Navy LCVP's using this type of system at times experience filling of the interior water passage of the engine with sand. Amphibians, therefore, are fitted with keel coolers located in the position best suited to the hull design.

Three feasible arrangements are:

1. A special recess in the hull with a grille protecting a compact heat exchanger element
2. Tubular or other heat exchanger, without protection, in a natural recess such as a wheel well or propeller tunnel
3. Pipes or special shapes welded or extruded with the bottom shell to provide heat exchange through the bottom

For the land mode, a truck-type radiator is fixed in a recess in the hull with a declutchable fan behind it. The recess is open on the top so that the air can be drawn through the radiator and exhausted. Tests run on amphibians have shown that land cooling must be sized to accommodate maximum engine output for long periods.

The major problems that plagued the LARC V and LARC XV cooling systems were:

a. Electrolytic corrosion of fan drives and heat exchangers

b. Insufficient cooling at high ambient temperatures

c. Magnetic fan clutch failures

7-10.5 CONTROL SYSTEMS

Control systems must be designed to be used by any Army driver in any environment in which the vehicle will operate. Some of the problems experienced in the development of recent amphibians have shown:

a. The need for a throttle locking device in the waterborne mode

b. That the orientation and direction of motion of controls should coincide with the direction of the action controlled

c. In general, the control wires and/or rods are too light and frequently bind, stretch, or bend requiring frequent adjustment and/or repair.

As an example of the complexity of these systems the controls and panel lights for a simple amphibian are listed by type in Table 7-47.

7-10.6 NAVIGATION SYSTEMS

For most operations, navigation of amphibians can be conducted without special devices because the route between the cargo ships and the landing beaches is normally within sighting distance. For those occasions when visibility is restricted, a compass and navigation lights are necessary safety features. A combination chartboard and navigation instrument is useful if a chart is used. It is also desirable to provide dry stowage for charts, maps, and manuals.

For those situations where the destination is not visible for part of the voyage, while following a single given compass course, a more sophisticated navigation system is required. One solution is to furnish, when necessary, a portable homing receiver for each vehicle to indicate to the operator the direction to a transmitter. The transmitters would also be portable and could be stationed on ships, the beach, or buoys. It is necessary that the vehicle crew be capable of setting the receiver to home on a number of different transmitters.

7-10.7 COMMUNICATION SYSTEMS

It is desirable to provide some form of communication between the control personnel and the vehicle crew. This can be accomplished with either portable or fixed equipment. In general, the same equipment provided for trucks will be satisfactory.

TABLE 7-47 LARC V CONTROLS

Start and Control Switches
Master switch
Ignition switch
Bilge pump switches
Converter lock switch
Horn button
Fan drive switch
Emergency override
Levers and Foot Pedals
Throttle
Forward-neutral-reverse shift lever
Land drive selector
Marine drive selector
Parking brake
Front wheel disconnect
Retarder pedal
Brake pedal
Accelerator
Light and Accessory Switches
Windshield wiper
Panel lights
Driving lights
Navigation lights
Heater controls
Defroster controls
Emergency Equipment Controls
Engine fire control handle
Manual bilge pump
Emergency tiller
Instrument and Warning Lights
Fuel gages
Ammeter
Engine and pressure
Engine water temperature
Fan stopped light
Speedometer
Low oil pressure light

7-10.8 VENTILATION SYSTEM

The air is changed in the enclosed sections of an amphibian for:

a. Engine combustion air requirements
b. Temperature control
c. Removal of noxious gases and resupply of breathing air
d. Removal of fuel vapors in machinery compartment
e. Engine cooling for air cooled engines, e.g., gas turbines

Except for air cooling of engines, the quantities of air involved are relatively small. The requirement is the design of an intake and exhaust system which will allow the air to pass but still not allow dangerous amounts of water into the hull in any reasonable operating condition. In general, no conditioning of the air is required except for possibly heating the air for the cab and passenger spaces.

In the presence of salt-spray, gas turbines salt up causing excessively high operating temperature and accelerated corrosion. This condition reduces the power level until the salt is washed off with fresh water. For this reason, the ventilation system for a gas turbine installation usually includes demisters, and filters. It is recommended that the engine manufacturer be consulted regarding the type and amount of filtering required for any specific design.

For very cold weather, it is desirable to provide for restricting the ventilation in the machinery space so that the temperature inside will stay above freezing and prevent ice from forming in the recesses and on the deck.

The cab should be provided with a fresh air supply, heated when necessary, and a good defroster. A change of air every two minutes is desirable for occupied spaces. For hot weather, devices should be available to secure some of the doors, windows, and hatches in an open position. A hatch in the cab roof is particularly useful for venting the hottest part of the cab. A fan to circulate the air in the engine space is beneficial to maintenance personnel.

7-10.9 LUBRICATION SYSTEM

There is such a large assortment of moving parts on an amphibian that the job of keeping all parts properly lubricated is critical and time-consuming. It is desirable to reduce the number of different lubricants to a minimum. It is also important to eliminate maintenance under the vehicle and in the bilge during the time between maintenance periods. On the LARC V and LARC XV, for example, the

propeller shaft bearing is to be greased every four hours of operation. For a continuous operation, this requires that the crew crawl under the vehicle while it is still wet and, possibly, while it is being loaded. It is understandable that this preventive maintenance is sometimes overlooked. The universal joints hidden away in the wet and dirty bilge suffer from the same problem.

The lubricating oil sump on engines built for land vehicles is sometimes not designed for the high angles of roll that occur on amphibians. The entire range of practical motions must be considered for the design of the lubricating oil system. Sump drains must be specially considered because the engines are usually mounted as low in the vehicle as possible. In the LARC V, for example, the engine oil was first drained into the bilge and then pumped into a waste container. This is obviously an inefficient procedure which increases maintenance costs.

Elements of a typical lubrication oil system include pumps, filters, heat exchangers, valves, piping, sumps, and level and pressure gages. The engine oil sump is not cooled by the air flow generated by the motion of the vehicle, hence a heat exchanger with sufficient capacity is required.

7-10.10 FRESH WATER SYSTEM

Fresh water systems are normally fitted on amphibians only for water washing gas turbines to eliminate the salt water. The requirements for the service should be obtained from the engine manufacturer.

7-10.11 COMPRESSED AIR SYSTEM

Compressed air may be used for engine starting, tire inflation, air powered tools, and control power, e.g., air brakes. The importance of variable tire inflation with regard to mobility is discussed in Ref. 26.

A compressed air system is relatively simple and consists of a compressor, accumulator piping, valves, and water traps.

7-10.12 CARGO HANDLING SYSTEM

Cargo handling is usually accomplished in three phases:

1. Transfer of cargo from ship to vehicle
2. Vehicle transit to destination
3. Off-load cargo at the land destination point

At sea, the cargo is handled easily with the ship's crane but there are serious difficulties in safely landing and securing the cargo on the amphibian cargo deck while it is rolling, heaving, and pitching. The location of the tie-down pads is important since a lack of pads may limit some types of cargo to a partial load. The lashing arrangements must be capable of being easily and quickly attached and adjusted by one man wearing gloves.

It has been suggested that lashing arrangements, designed to fail if the amphibian rolls in excess of some fixed angle, might be a desirable safety feature.

Heavy swinging drafts of cargo present particular problems. The draft will damage any projection above and adjacent to the cargo area that is not properly designed.

The deck in the cargo area also receives severe punishment from heavy items of cargo with sharp corners. Some remedies are the provision of a layer of rubber bonded to the deck, a layer of plywood, or increased deck plating scantlings. The last solution probably requires the least maintenance.

The transit phase is not a particular problem with regard to the cargo handling system unless the cargo shifts or must be jettisoned. Fabric side curtains supported with wire rope prevent shipping some water in the cargo space and still allow the water trapped on the cargo deck to drain overboard quickly.

The protection of cargo from the weather with a tarpaulin can be useful. The covering and supporting structure must be strong enough to withstand effects of the surf. Configurations with nonwatertight cargo decks must be protected to prevent swamping. Protected storage for all portable equipment must be provided.

The last phase, off-loading the cargo, is a problem because of the great variety of the equipment available to handle the heavy items. A flush deck arrangement with side curtains allows the use of fork lift trucks. Cargo that must be lifted out of a well is particularly difficult to handle. The provision of self-contained unloading gear requires additional weight and space, and penalizes the performance of the vehicle. Ramps can be used for wheeled

cargo and for rolling off heavy boxes. This is a slow method, however, and requires added weight, increased complexity, and maintenance. A winch with a flexible arrangement of snatch blocks, deck pads, wire rope, nylon line, and assorted fittings can be used in a variety of ways to assist in off-loading in many circumstances.

7-10.13 FIRE FIGHTING SYSTEM

Combustible materials on an amphibian include fuel, electrical equipment, cargo, and possibly the hull of a Fiberglas vehicle. The vehicles are relatively compact and easily disabled by fire so it is necessary that the crew be able to detect and extinguish a fire quickly.

The fire fighting system may include a detection device, and portable and fixed extinguishers with the necessary controls. There are a number of types of detection devices that might be used. The type selected should be quick-acting, repeating, accurate, and reliable. The detector may actuate lights, alarms, and a fixed extinguishing system.

Either carbon dioxide or monobromotrifluoromethane Halon 1301 may be used as the extinguishing agent. The latter is reportedly much more effective than CO_2 and is only a little more expensive. Portable extinguishers should be located in the cab and also at the aft end of the cargo compartment so that at least one is accessible in any situation.

A fixed extinguishing system has a supply of extinguishing agent, distribution system, nozzles, and local or remote control and is normally installed in the engine space. In order for the system to be safe and effective it must:

a. Contain sufficient agent to fill the compartment where the fire is to be extinguished

b. Permit evacuation of personnel from the space

c. Stop the fuel supply

If the engine draws combustion air from the space, it will stop as soon as the agent drives the air from the compartment. The control may be manual or automatic, and local or remote. The control mechanism should be relatively fireproof so that it is not damaged by fire before the system is actuated. If local control is used, it should be located where it can be reached during a fire. Ref. 115 recommends quantities of CO_2 to be installed in fixed systems on boats.

7-10.14 BILGE PUMPING SYSTEM

The primary consideration for the design of the bilge pumping system is the rate at which water can enter the bilge. If the vehicle has a cargo well deck that is open to the bilge, similar to the DUKW arrangement, the safety of the vehicle in the surf or high seas depends on the pumps' removing the water faster than the average flooding rate. Portable covers such as tarpaulins supported by bow frames are very effective in reducing the quantity of water. The covers must be sufficiently strong to prevent failure in the surf. The DUKW's were equipped with tarpaulin covers and, on occasion, they were swamped when the covers were split by the force of the plunging surf.

For vehicles with a watertight deck, the pump must be able to handle the water that can come aboard through the ventilation system, exhaust system, hatches, and, possibly, the machinery cooling system waterpump to allow for a broken water line. The pump capacity should also be greater than the possible inflow through the hull drains so that the vehicle will not founder if the drains are accidentally left open.

When the amphibian is receiving cargo from a ship in rough weather, the amount of water entering the hull may be relatively high due to the interacting wave patterns between the ship and the vehicle. Usually the amphibian will be operating with a moderate amount of ahead thrust to maintain tension on the spring line and, consequently, the engine will probably be operating at a low rpm. The bilge pump system design must provide sufficient pumping capacity at low engine power levels to prevent the accumulation of an excessive amount of bilge water in this situation. Table 7-48 lists bilge pumping rates for amphibians.

Other design requirements include:

a. The pump suctions must be located where the water will collect in any condition. Some craft are lowest forward while at rest and lowest aft underway. If the cargo is off center, the low point will be on one side for vehicles with flat bottoms.

b. Pumps should be self-priming and designed to run dry continuously.

c. Operation can be controlled automatically or manually. The overboard discharge should be located above the waterline where the operator can observe the discharge. A high water alarm

TABLE 7-48 BILGE PUMP RATES

Vehicle	Power Source	Number of Pumps	Capacity of Each Pump	Total Capacity, GPM
DUKW	Mechanical	1	225 GPM	> 300
	Mechanical	1	> 75 GPM	
	Manual	1	25 GPM	
LARC V	Hydraulic	3	300 GPM	900
	Manual	1	0.75 gallon per stroke	
LARC XV	Hydraulic	3	300 GPM	900
	Manual	2	0.75 gallon per stroke	
LARC LX	Hydraulic	3	100 GPM (bilge)	300
	Hydraulic	2	2000 GPM (cargo well)	4000

can also be provided.

d. Hydraulic drive or submersible electric drive should be used so that high water level in the bilge does not stop the pump.

e. The suctions should be located to be observed during normal maintenance so that rags and other debris that collect at the pump suction can be removed as they collect. The suction should also be accessible when there is at least a one foot depth of water in the bilge. This criterion has been overlooked in boats which have foundered only because the crew could not clear the pump intakes.

f. Bottom drains are normally fitted to keep the hull dry inside during operation on land. It is important that they be designed so that parts are not easily lost, and the drains can be easily removed and reinstalled.

g. The system piping should be designed to withstand the maximum pressure that the pump can provide.

h. The bilge suctions should be installed to provide a free flow area through the strainer at least 1.5 times the suction pipe cross-sectional area.

7-10.15 MOORING AND ANCHORING SYSTEMS

Amphibians are not expected to anchor frequently or for long periods. Accordingly, a lightweight anchor, a short length of anchor chain, and a nylon line are sufficient for the anchoring system.

The present system for holding an amphibian alongside a ship while receiving cargo provides for the ship to furnish a spring line which is attached at the forward inboard corner of the vehicle and led aft to the attachment on the ship. The amphibian is then held against the ship by running the propulsor in the slow ahead condition with the steering device directed to hold the stern against the ship's side.

The mooring system consists of properly placed cleats and nylon mooring lines. The lifting and/or towing pads can be used if they are in the correct position. Ref. 115 may be used for guidance in the selection of components for the mooring and anchoring systems.

7-10.16 EXHAUST SYSTEM

The exhaust systems used for amphibians are similar to those used on trucks, i.e., dry mufflers, and they exhaust out of the top of the vehicle. The entire system must be shielded or insulated for personnel safety. The system must be arranged so that engine motion, thermal expansion, and local hot spots do not crack the exhaust lines. The exhaust should not discharge forward of the operator's position.

Gas turbine exhausts are hot, large, and noisy. The engine manufacturer should be consulted to determine the requirements for a particular engine. The Navy conducts research on this problem and the Boat Engineering Department,

Norfolk Division of the Naval Ship Engineering Center, Norfolk, Virginia, may be contacted for the latest information.

7-10.17 SAFETY AND LIFESAVING SYSTEMS

Safety has to be discussed throughout the text because it must be considered with each part of the design. Some of the more important features required are:

a. Fast, accessible escape routes from the cab in any situation including a capsized vehicle
b. Handholds so that the entire vehicle is accessible at sea in an emergency
c. Convenient life rings or lifesaving devices
d. Emergency tiller
e. Fire fighting equipment (par. 7-10.13)

7-10.18 HEATING AND DEFROSTING SYSTEMS

Heating and defrosting systems can be provided with hot water from the engine cooling system or require a separately fired burner. The latter should use the same fuel as the engines. Hot water heaters are not as efficient as separately fired heaters but require less maintenance. The heat must be properly distributed and free of fumes. The defroster should be capable of blowing fresh, unheated air, independent of the heater.

7-10.19 ON EQUIPMENT MATERIEL (OEM)

OEM are portable items of basic issue placed on board vehicles, as necessary for proper operation and maintenance by the crew, including:

a. Tool kit
b. Boat hook
c. Boarding ladder (if not built-in)
d. First aid kit
e. Anchor and line
f. Repair parts
g. Portable fire extinguisher
h. Life preservers
i. Mooring lines
j. Lifesaving ring buoy
k. Spotlight/flashlight
l. Lubrication order
m. Technical manual
n. Marine fenders
o. Pyrotechnic pistol
p. Radio set
q. Signal lamp
r. Portable trouble light
s. Emergency steering wrench

Proper storage must be provided for all OEM and some personal equipment for the crew, including such items as coats, boots, hats, helmets, rifles, ammunition, tool boxes, and charts.

7-10.20 COLD WEATHER KIT

Operating in very cold weather is a severe condition for any vehicle. If an air heater is used in the bilge of an amphibian and the air is circulated throughout the vehicle while the fresh air ventilation is restricted, many of the problems are modified. The procedure warms up all of the interior components and reduces the amount of external icing on the hull. When securing the vehicle, any seawater entrained in the hull or machinery must be removed.

Engine starting problems are the same for amphibians as they are for trucks.

SECTION VI COST ESTIMATING

7-11 COST ESTIMATING AND THE OVERALL DESIGN PROCESS

The financial and technical resources allotted to our national defense are inadequate to develop and produce all of the possible weapons and logistical systems that would assure continued superiority in our defense posture. As a result, a system analysis approach has been initiated to permit rational selection between alternative systems so that the greatest return for a fixed expenditure will be assured. With this approach, certain decisions will have been made long before a specific vehicle design is begun, i.e.:

a. How the vehicle concept fits into the total plan for national defense, thus establishing performance requirements
b. How the vehicle will be transported, supplied, maintained, and manned as a

subsystem in the total integrated logistic support plan, thus defining support requirements

c. What expenditure will be allocated to the system based on its total life cycle cost, thus defining budget requirements

Any given vehicle, despite its actual excellent physical performance, will not represent an effective solution to the total defense problem if it cannot be developed within an established budget. Cost estimating performs two vital functions in the development of a new amphibian design. The first is to provide the system analyst or planner an appraisal of the amount of performance and capability that can be reasonably expected for a given expenditure. The second function is the provision of the necessary tools to permit the designer to control costs so that the system will be developed, constructed, and operated within the allowed budget. The discussion that follows will consider cost estimating methods that are useful to the designer. The methods used by the system analyst in the planning stages are complex and beyond the scope of this handbook.

7-11.1 THE TOTAL, OR LIFE CYCLE, COST CONCEPT

To illustrate the concept of total life cycle cost, it is useful to consider the actual costs that would be incurred by a change in design of a small fitting to be made of a new material. Such costs might include:

a. The cost of research to study the properties of the new material

b. The development cost of adapting the new material to the specific requirements of the fitting

c. The cost of redesigning the particular fitting and any corresponding changes that become necessary in the overall design

d. The cost of buying the particular part including purchase price, tooling charges, and possible costs due to long lead times, delays, etc.

e. The cost of testing necessary to verify the performance of the new fitting and the systems of which it is a part, including alterations or replacements needed to make the new fitting acceptable

f. The cost of operating, maintaining, and repairing the fitting including any special training needed to operate an unusual system, and costs for shipping and storing sufficient inventories of replacement parts

While this example is directed toward a small part of a vehicle, it can be seen that it would apply to changes in subsystems such as alternative power plants or to total vehicle concepts. Thus, life cycle cost is the total cost of a system over the entire lifetime of the vehicle.

7-11.2 COST ESTIMATING METHOD

The following two principles are basic to the philosophy of cost estimating:

1. The higher the degree to which the cost estimate is broken down into types of costs and divisions of systems into subsystems and individual parts, the more reliable the estimate should be. This follows because the costs of different systems are characterized by different, but predictable, rules. For example, the cost of constructing an aluminum hull structure can probably be expressed as a function of hull weight in dollars per pound, whereas fuel costs are related to engine size and time in operation in terms of dollars per horsepower-hour.

2. Cost estimating is largely based on past experience. Hence, the more complete, accurate, and detailed the past cost figures are, the more reliable the future cost estimate should be. The historical cost data figures may be adjusted to allow for known changes such as inflation. Where undefined areas exist, the designer must use good judgment in modifying estimating methods until he is satisfied that it represents a reasonable approach.

It should be emphasized that high cost estimates by the designer in order to prevent apparent cost growth is not a responsible approach to cost estimating. Conversely, unreasonably low estimates must be considered unresponsive.

Cost estimating of individual systems or components are treated in the paragraphs which follow.

7-11.3 RESEARCH AND DEVELOPMENT COSTS

Research includes the studies, experiments, and calculations needed to develop and verify new solutions to the basic engineering problems encountered during the evolution of an amphibian design. Development is the process whereby the results of research and advancing technology are integrated with the vehicle performance requirements to establish a

satisfactory design configuration. Included in this process are the construction, testing, and evaluation of new prototype subsystems, systems, and complete vehicles. The process may be continued through change orders and product improvement studies.

Research and development (R & D) costs are probably the most difficult to estimate because every new vehicle concept will present new problem areas requiring R & D. Since the solution is not known, it is difficult to estimate how much cost will be involved in arriving at a solution. However, R & D costs are typically a very small part of the vehicle total life cycle cost. Hence, accuracy in estimating this part is not critical to the accuracy of the total estimate.

Before a cost estimate can be made, the problem areas must be recognized and defined. This requires a thorough study of the vehicle performance requirements in light of current technology. R & D costs are directly related to the number and magnitude of such problems that require solution. The further removed from past vehicles and the more difficult the performance requirements, the higher the cost for R & D. A thorough understanding of the technological problem areas and insight concerning the probable solutions, coupled with past experience and data, are all necessary to make reasonable cost estimates of R & D.

7-11.4 DESIGN COSTS

Design costs are incurred in transforming the vehicle concept and performance requirements into specific written instructions (plan and specifications) from which the final product can be built. Sometimes termed engineering costs, design costs can be derived from historical data:

a. For a given type of vehicle, the design labor may be expressed as a constant ratio of direct construction labor for one vehicle. The design material (drawings and specifications) is, in turn, proportional to design labor. Thus, the estimate of design costs may be made by adding factors for design labor and design material and multiplying by the projected number of construction labor hours.

b. If no similar vehicle has been constructed, an alternative method would be to estimate the average cost of preparing an engineering drawing and a page of written specifications, based on experience. These cost figures would then be multiplied by projected numbers of drawings and pages of specifications, again based on experience.

In either case, allowances should be made for contingencies such as travel expense, and cost of evaluating new equipment and manufacturers' proposals if they differ from the general conditions upon which the historical data were based.

The design costs are likely to be divided between the designer and the builders, where these are different agencies. Care should be exercised in allocating this cost since the builder's portion will show up as construction costs and will depend on the share of the design work furnished by the builder.

It should be noted that the total cost of R & D and design will be only a small percentage of the total life cycle cost for any vehicle produced in large quantities. Research and development costs for new aircraft designs range from 1 to 5 percent of life cycle cost. Aircraft are very complex systems and evolve from intensive R & D efforts. Research, development, and design efforts for amphibians are small by comparison. For example, the LARC V and LARC XV designs were each bought for less than $250,000.

7-11.5 CONSTRUCTION COSTS

Total construction costs may be considered to include:

a. Initial contract estimate

b. Cost of Government-furnished material (GFM)

c. Cost for change orders

d. Cost for inspection and administration

e. Cost of delays and unforeseen technical problems

f. Cost of increased labor rates and material prices over long construction periods.

7-11.5.1 Estimating Methods

Various designers have their proprietary methods for estimating construction costs. The method given in the paragraphs which follow represents one approach to the problem. The vehicle components are catalogued into suitable cost groups. These are the same groups used in weight estimating as given in Table 6-6:

A. Hull Structure

B. Machinery
C. Main Drive Train
D. Marine Drive Train
E. Running Gear and Suspension System
F. Miscellaneous Systems

Two additional groups should be included:

G. Engineering and Procurement—includes the cost of engineering supervision in addition to design work to be performed by the builder and the cost of purchasing components

H. Construction Services—jigs, fixtures, performance bonds, and insurance, to be charged against the specific contract.

Group A, Hull Structure, costs can be characterized by a price per pound for materials and a number of man-hours per pound for construction labor. These numbers are based on historical data. The same technique is used for the items in Groups B to F.

Group G, Engineering and Procurement, can be expressed as a function of the total labor in groups B to F above. Group G materials are a fixed cost per man-hour of Group G labor. Group H labor and material, likewise, are functions of the total labor of Groups B to F.

The sum of labor in Groups A to H is adjusted by a multiplier to reflect overhead and is priced at an average labor rate. The total labor cost is added to the total material cost and then multiplied by a suitable factor for the builder's profit.

The above discussion shows some of the complexity of the cost estimating process. More important, it demonstrates the reliance on previous cost data. Available cost records must be accurate and clearly understood with respect to category groupings. Unfortunately, cost data for existing amphibians are not readily available. Hopefully, more attention to cost accounting will be given for future amphibians and will be available to the designer.

7-11.5.2 Multiple Construction Savings

The cost of succeeding duplicate vehicles should be less than the first vehicle because of the following savings that result from duplication:

a. Nonrecurring charges, including costs of molds, patterns, jigs, and fixtures

b. Vendors' savings, such as quantity price discounts from manufacturers and suppliers

c. Labor savings, due to increased efficiency for succeeding vehicles

To estimate the effects of these savings it is customary to:

a. Estimate the total of nonrecurring changes, adjusted for such items as mold repairs, jig and fixture replacement, etc. This total is divided by the number of vehicles contemplated to obtain a cost per unit.

b. Obtain quotations from manufacturers regarding quantity price discounts to be expected on major vendor items.

c. From past experience with labor for multiple vehicles, plot a curve of hours expended on each subsequent vehicle, expressed as a percentage of the first. This "learning curve" can then be applied to the hours estimated for the first vehicle to determine the hours spent on subsequent vehicles. Fig. 7-107 shows a typical learning curve.

7-11.5.3 Manufacturer's Estimates

In order to confirm cost estimates it is necessary to obtain quotations from a representative number of suppliers. Two distinct types of estimates are common in industry and are considered here:

1. Budget Prices. These figures are requested from manufacturers in order to ascertain the price range of certain services. Prepared as a courtesy by manufacturers, these figures are usually very approximate and obligate neither party. In asking for budget estimates the designer should realize that they are provided out of courtesy, and care should be exercised to avoid taking undue advantage of manufacturers.

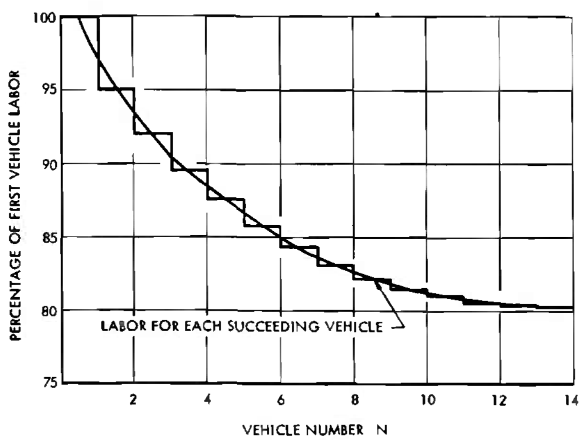


Figure 7-107. Typical Learning Curve

It is also important to ask the manufacturer to state weights, probable delivery schedules, and to include allowances for crating, loading, and freight.

2. Quotations. These represent offers on the part of the manufacturer to supply an exact item by a specified delivery date, satisfying certain performance and construction requirements. Usually the quoted price is firm for a limited time, such as 90 days from date of quote. This type of response represents a legal commitment on the part of the supplier and is prepared after detailed estimates.

7-11.5.4 Current Costs

Current (1968) catalog prices for the LARC series of amphibians are given here for information:

LARC V $ 44,207 each, for 250 vehicles
LARC XV $165,000 each, for 110-115 vehicles
LARC LX $390,000 each, for 10 vehicles

7-11.6 TESTING COSTS

A certain amount of the testing of any vehicle is usually conducted by the builder to demonstrate compliance with the vehicle specifications (proof tests). This testing cost is included in the contract cost and is not of concern to the designer as a separate item.

Costs associated with operational tests (from R & D to acceptance tests) conducted by the Army are of the following types:

a. Repairs and alterations to vehicle

b. Fuel and lubricants

c. Wages for operating personnel and consultants

d. Transportation charges for operating personnel and materials to and from test site

e. Overhead associated with the test facility

Historical cost data in this category should be sufficiently well documented for cost estimates, provided the number of tests and their probable duration (allowing for delays for repairs and inclement weather) can be estimated.

7-11.7 OPERATION, MAINTENANCE, AND REPAIR COSTS

Operation, maintenance, and repair costs are major items in the total life cycle cost. These costs include:

a. Cost of materials (fuel, lubricant, etc.) consumed during the life of the vehicle

b. Wages paid to operators and support personnel

c. Cost of special training necessary to operate and maintain vehicle

d. Cost for repairs including labor and parts

The largest portion of these costs is expended in wages to operate and support the vehicle. The wages for operating personnel are easily estimated from the number of crew that are needed to operate the vehicle. However, the wages for support personnel needed to maintain the vehicle are a function of the maintenance requirement which, in turn, depends on how much effort is directed toward improved maintenance in the development and design stages of the amphibian life cycle. Table 7-49 presents maintenance data reported by field units in Southeast Asia for the LARC V and LARC LX from The Army Equipment Record System (TAERS) (TM 38-750), along with forecasts quoted from U. S. Army Mobility Equipment Command (MECOM) studies for comparison. Organizational maintenance includes the routine servicing, adjustment, inspection, and minor repairs that are performed by the vehicle user-operator. Support maintenance includes inspection, complicated adjustment, major repairs and modifications, and major replacement performed by support unit personnel who are better equipped and trained for more complex maintenance jobs. Maintenance is reported in terms of man hours of maintenance required per hour of vehicle operation.

TAERS records do not include organizational level maintenance. However, the MECOM studies may be considered accurate and representative of actual requirements with respect to organizational maintenance. The projected estimates of support maintenance were on the high side, almost twice the actual man hours reported in the case of the LARC V. As a result, the estimated total maintenance for this vehicle was approximately 25 percent greater than the actual maintenance.

The other costs involved in operation, maintenance, and repair are best estimated using experience, records of similar vehicles, and good judgment.

TABLE 7-49 LARC AMPHIBIAN MAINTENANCE, MAN-HOURS/OPERATION HOUR

	LARC V	LARC XV	LARC LX
Organizational Maintenance (MECOM-projected)	0.46	0.58	1.48
Support Maintenance (MECOM-projected)	0.35	0.41	0.34
Support Maintenance (TAERS-recorded)	0.19	—	0.32

SECTION VII DETAIL DESIGN COORDINATION, SUMMARY, AND OUTPUT

7-12 DESIGN COORDINATION, SUMMARY, AND OUTPUT

Preparation of the detail design, or working plans, is normally accomplished with the prototype construction within a single contract. This provides for good communications between the designer and the builder. In addition to the main contract, there may be component or system subcontractors, research contractors, and contractors furnishing components directly to the Army. For the detail design to proceed smoothly in an orderly fashion, many segments of the design must be conducted simultaneously. This requires flexible planning and continuous communications.

7-12.1 SCHEDULING

Scheduling the detail design requires the gathering, at the proper time, of qualified designers and the necessary information. Ideally, the critical categories of information are determined and the availability dates are established. With these dates as a framework, each item of work can be scheduled. This, of course, is ideal and in practice there are many conflicts and gaps in the information available to the planners. One typical problem is the long period of time required to manufacture and test a waterborne model. The delay in obtaining these results places a greater importance on the initial resistance and propulsion estimates because a significant error in these estimates may cause major changes in the design close to the completion of the design.

7-12.2 DETAIL DESIGN PLANWORK

The plans and specifications must furnish sufficient information so that the materials can be ordered and fabricated by a competent manufacturer who is familiar with vehicle construction. Any necessary information omitted increases the construction cost. Military specifications for each class of vehicle contain lists of the applicable documents, including a list of required plans. This plan list for a similar vehicle may be used to estimate the type and number of drawings required for a new design.

7-12.3 FINAL DESIGN REPORT

The final design report is used by the vehicle development team and by the designers for their work in designing later generations of amphibians.

The final design report should contain:

a. Final vehicle description

b. Final vehicle performance data

c. Design history

d. Major design decisions

e. Summary of design techniques and assumptions

The major design decisions must be recorded periodically or the reasons for each decision will be lost.

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PART THREE EVALUATION AND TESTING OF WHEELED AMPHIBIANS

CHAPTER 8 EVALUATION AND TESTING

8-1 OBJECTIVE OF TESTS AND GENERAL DISCUSSION

For any given test there are two basic possible reasons for its conduct. One reason is to determine the characteristics of some unknown phenomenon or to determine a not readily predictable result. A second basic purpose of testing, in general, is to verify (or disprove) an expected or previously predicted result.

The purpose or objective of a test is, therefore, usually related to the stage of development of the test item. For example, research or feasibility tests fall into the area of testing where the test results indicate whether a concept is feasible or attainable. On the other hand, tests are performed on prototype amphibians and other Army materiel to determine whether a predicted or designed characteristic or result has actually been attained.

This chapter will deal mainly with the latter type testing where the feasibility of a concept or vehicle has been proven or accepted, and the interest of further tests is to verify (or disprove) that the requirements (i.e., the QMR's) of a design have been met. This limitation has been made for two reasons. First, a complete dissertation on testing and testing philosophy would require a Handbook in itself and, second, the maximum benefit to the designer can be realized by a knowledge of the amphibian tests which will eventually lead to an item's acceptance into Army service.

This chapter is of value to the amphibian designer, but is no substitute for a thorough knowledge of the U.S. Army philosophy of research and testing. This entire picture is a necessary tool for the program manager, project manager, and planning personnel involved in an amphibian design and, of course, is mandatory knowledge for the designer. Ref. 1 presents a good basic description of the Army test organizations, testing concepts, testing documents, testing terms and definitions, funding, etc., along with pertinent references from which greater detail in any specific area of interest may be obtained.

The primary objective of Army materiel testing as stated in Ref. 1 is "to insure that materiel for Army use satisfies the minimum operational, technical, and safety requirements in all types of environments of intended use as described in the QMR.... or equivalent requirements, and to determine what changes are required to make new materiel suitable and safe for Army use".

What this means to the designer of a wheeled amphibian is that the (usual) ultimate goal of a vehicle design is to become "Type Classified" in accordance with the design objectives and QMR.

Army materiel is Type Classified in order to associate the particular item of materiel with regard to development or service status relative to other Army items. The levels of Type Classification and description are as follows:

a. Development Type. Materiel (amphibian) being developed or tested to meet a QMR.

b. Limited Production Type. Materiel in limited production that has not yet passed all tests required; however, urgency of the situation warrants initiation of production and the confidence level of meeting minimum requirements is high.

c. Standard Type. Materiel which has completed all tests satisfactorily; is authorized for Army use; and is included in published adopted item lists.

d. Contingency and Training Type. Materiel which is not acceptable for U. S. Army operational requirements, but may be used for training or to fill a need until a standard (type) item becomes available.

e. Obsolete Type. Materiel which is no longer acceptable for U.S. Army use and is being or has been phased out of the supply system.

The methods and requirements for type classification of Army materiel are presented in detail in Ref. 1, pages II-12 through II-15.

There is another, sometimes overlooked, objective of testing which is of major importance to the designer. That is the

"feedback" to the designer of quantitative test data which can be compared with the performance predicted during the design and development phases. In this manner, the design methods and predictions can be modified to provide more accurate or more quantitative design predictions for future vehicle developments.

From the long range viewpoint, this feedback of test data to the designer is of equal or greater importance than the short range objective of vehicle type classification since it is these data which, in part, should prevent repetition of design errors and lay the groundwork for further improvement of amphibian designs.

Acquisition of test data is the responsibility of the development command (designer). Each development program organization should establish channels for the procurement of test data. Methods of obtaining data and its presentation should be specified so that the test results are in a useful form (comparable with other amphibians or design data). The paragraphs which follow are intended to aid the designer in establishing a meaningful test program including test specifications, conduct of tests, and data formats which fall under his cognizance.

8-2 TYPES OF TESTS

According to the definitions or doctrines of the U.S. Army Test and Evaluation Command, (USATECOM), there are three broad classifications of testing: Category I Tests, Category II Tests, and Special Tests.

Category I Tests are those in which USATECOM is responsible for the establishment of the test objectives, preparation and approval of the plan of test, and processing and distribution of the report of test. The basic philosophy of Category I Tests is the provision of timely, objective, and factual test results to HQ, AMC, in order that decisions affecting type classification include materiel performance considerations. These results are obtained by independent testing and evaluation of the item, and normally lead to type classification of the materiel undergoing tests. USATECOM has sole primary responsibility for the planning and conducting of Engineering Tests, Service Tests, Integrated Engineering Service Tests, Environmental Tests, Confirmatory Tests (Type I), and Check Tests.

Category II Tests are those in which USATECOM is performing a testing service for the requesting agency and in which the test objectives, plan of tests, and processing and distribution of the report of test are the responsibility of the requester. The concept of Category II Tests is the providing of USATECOM facilities and knowledge to using commands and agencies in order to expedite the conduct of development and procurement test programs.

The above definitions were extracted from Ref. 1, and are of value in attempting to establish a logical sequence of testing and a test program. The significant inference in these definitions is that the Development Command or Agency has no control over the Category I Tests on an amphibian. Category I Tests are the sole responsibility of USATECOM and are developed directly from the amphibian QMR. These Category I Tests, alone, determine whether the vehicle has been successfully designed to meet the requirements for Type Classification.

Category II Tests are the responsibility of the Developing Agency or Command, and therefore are directly influenced by the designer. In this regard, it is important that the Technical Characteristics (TC) developed by the designers from the QMR's are based on the interpretation of the QMR made by the USATECOM in conjunction with the committee composed of personnel from the Development Command or Agencies which generated the QMR. This is important since the tests undertaken by the development agencies, whether performed by USATECOM or not, will most likely refer to the TC as opposed to the QMR; however, the final amphibian design will be required to pass the tests prescribed by USATECOM, which are strictly based on interpretation of the QMR. This point is discussed further in par. 8-3.

8-2.1 CATEGORY I TESTS

The following are Category I Tests which are conducted on a vehicle after it is released to USATECOM by the Development Command(s) for Type Classification. It should be noted that all Category I Tests are intended to verify (or disprove) that an amphibian meets the QMR.

8-2.1.1 Engineering Tests

The purpose of Engineering Tests is to determine to what degree the test item meets the performance characteristics expressed in the QMR and whether the item is safe for use by troops during Service Tests. The basic concept of Engineering Tests is the elimination of human bias and errors in judgment so far as possible through the use of engineering and scientific methods of testing.

The results of the Engineering Tests should provide sufficient information to permit determination of:

a. Degree to which the technical performance of the item meets the requirements of the QMR

b. Relative safety of the item in the hands of Army troops

c. Suitability of the item for conduct of Service Tests

8-2.1.2 Service Tests

The purpose of Service Tests is to determine to what degree the test item meets the Military Characteristics expressed in the QMR, and the suitability of the item and the maintenance package for use by the Army. During the various tests that comprise a complete Service Test Program, factual records are maintained showing the item's performance under specific conditions. This record must be complete and explicit since it forms the primary basis for decisions by higher headquarters on the merits of the tested item. Except in the most unusual circumstances, the test results should provide sufficient information to permit determination of the degree to which the item meets the current military requirements of the prospective user.

Service Tests use subjective approaches to produce test results. The tests prescribed are conducted from the user standpoint under simulated or actual field conditions.

8-2.1.3 Integrated Engineering Service Tests

The purpose of an Integrated Engineering Service Test is to minimize testing time and cost by running Engineering and Service Tests at one location. Complete integration or partial integration constitutes an integrated test.

8-2.1.4 Environmental Tests

Environmental Tests are performed to determine if an item performs effectively in the environments of its intended use. Environmental Tests form an integral part of testing, at all phases, whether it be Engineer Design Tests, Engineering Tests, Service Tests, or Troop Tests. Testing in simulated climatic extremes is normally used to the maximum extent in the Engineer Design and Engineering test phases. Testing in extreme natural climatic environments is used to substantiate or supplement data obtained from simulated tests. Category I testing in natural environments will normally be an Integrated Engineering Service Test. Since testing in natural locations of extreme climate is costly in terms of manpower, money, materiel, and time; each individual item's requirements for testing in natural environments must be carefully determined before submitting the item for such testing. Such predetermination of necessary testing in natural environments is vitally important and should not be omitted in test design.

8-2.1.5 Confirmatory Tests (Type I)

A Confirmatory Test, Type I, is the test or investigation conducted on an item or system, after type classification as standard or limited production, using early production models to insure that required modifications not previously tested are acceptable and to insure that the modifications have not adversely affected the Technical and Military Characteristics of the item or system. Service Test procedures are utilized to insure that no deficiencies or shortcomings have been created.

8-2.1.6 Check Tests

The purpose of Check Tests is to determine on Service Test models of items whether major deficiencies found in the Service Test have been corrected, these deficiencies being of such a nature that the item was found unsuitable for type classification. The philosophy of Check Tests is to retest items using the Service Test procedures to insure that previous deficiencies and shortcomings have been eliminated, and no new ones have been created.

8-2.2 CATEGORY II TESTS

Category II Tests are the responsibility of the Development Command and are conducted by anyone selected by that Command (i.e., private contractor, USATECOM, or the Development Command).

8-2.2.1 Research Tests

Tests conducted during the research phase in order to confirm concepts and to further research projects and tasks.

8-2.2.2 Feasibility Tests

The determination of a process of technical examination and study of the possibility of attainment of end-item materiel development. Technical feasibility consists of two parts:

1. Long range or state-of-the-art study wherein the probability of attaining general technological goals is determined
2. Feasibility study of a desired end-item after the Military Characteristics are known

8-2.2.3 Component Engineering Tests

Engineering-type tests are conducted by or under the direction of the developing agency where the objective is the development of component design concepts, or the evaluation of components embodying new and advanced state-of-the-art principles. The test item need not be specifically related to any existing end-item development.

8-2.2.4 Engineer Design Tests

These tests are conducted to determine inherent structural, electrical, and other physical and chemical properties of materials and items, including the effects of environmental stresses on these properties. These tests collect design data; confirm preliminary concepts and calculations; and determine the compatibility of the components. For complex systems, e.g., an amphibian system, these tests may include a complete system demonstration. The Engineer Design Tests will be formulated by the developer after approval of the design concept and prior to the initiation of prototype fabrication. Applicable Engineer Design Test data will be made available by the developing agency for use in Engineering and Service Tests.

Engineer Design testing is primarily for the purpose of obtaining data necessary for initial design of the prototype item, or specific components thereof, whereas determining the performance characteristics of the prototype is the purpose of the Engineering Test. In the interest of expediting testing and completion of development, requirements for design data for further refinement and completion of the development can and should be provided by the Engineer Design Test, including Research and Development Acceptance testing of a contract item, particularly when complex elements of design or development are involved.

8-2.2.5 Research and Development Acceptance Tests

Research and Development (R&D) Acceptance Tests are conducted on an item or system designed and developed by a contractor to insure that the specifications of the development contract have been fulfilled. Acceptance of the item or system for Engineering and Service Tests is dependent on the results of the R&D Acceptance Test. The effectiveness, or readiness, of a given development project, however, is measured at the time of type classification, not at the time it begins Engineering and Service Tests. Early initiation of Engineering and Service Tests, irrespective of minor deficiencies anticipated in these tests, as a means of expediting an overall development process should be the goal of both developing and testing agencies.

8-2.2.6 Product Improvement Tests

Product Improvement Tests are conducted on modified standard items of Army materiel to verify that essential Military Characteristics have not been adversely affected and to establish the durability, operational capability, and maintainability of the modified item.

8-2.2.7 Preproduction Tests

Preproduction Tests are Engineering-type tests of a preproduction model or initial production sample—produced in accordance with the supply procurement specifications and drawings using the same methods, material, and

equipment as will be used during regular production—to verify production drawings, processes, and materials.

8-2.2.8 Initial Production Tests

Initial Production Tests are tests of an early item or system from the first production run which are used to verify the adequacy and quality of the materiel when manufactured according to the production drawings and of the mass production processes.

8-2.2.9 Comparison Tests

Comparison Tests of random samples of production line items are conducted as a quality assurance measure in order to detect any design, manufacturing, or inspection deficiencies that may reduce the effective operation of the items by the using agency.

8-2.2.10 Acceptance Tests

Acceptance Tests are conducted on individual products, or lots of materiel items, which have been submitted for acceptance to determine conformance of the products or lots with requirements and specifications prior to acceptance.

8-2.2.11 Reconditioning Tests

Reconditioning Tests are conducted on standard items, which have been modified to correct deficiencies discovered during use, to verify that essential Military Characteristics have not been adversely affected, and to establish the durability, operational capability, and maintainability of the modified item.

8-2.3 SPECIAL TESTS

Special Tests are so named and categorized because of their unusual nature in regard to reporting and control procedures, method of conduct, facilities, and services required. Special Tests which are generally applicable to wheeled logistic amphibians are defined in the paragraphs which follow.

8-2.3.1 Confirmatory Tests (Type II)

Type II Confirmatory Tests are used to evaluate a selected item or system of early production materiel, after type classification as Standard or Limited Production, by expedited and intensive testing in the field to preclude time-consuming retrofit programs. Items may be recommended for Type II Confirmatory Testing by project managers, commodity commands, separate installations and activities reporting directly to AMC Headquarters, and the directorates of AMC Headquarters. DCSLOG, in coordination with the other members of the Army General Staff, acts on recommendations of the major developing agencies in the selection of items and furnishes the major developing agencies an approved list of items to undergo Type II Confirmatory Tests. Service Test procedures are utilized to insure that no deficiencies or shortcomings have been created.

8-2.3.2 Surveillance Tests

Surveillance Testing is performed on items that are subject to rapid deterioration of materiel or performance. These items are stored and tested on a continuing basis in all climatic environmental conditions to determine the extent of deterioration as compared with the original requirements. Facilities and test personnel of the commodity commands, separate installations, and activities reporting directly to USAMC as well as USATECOM test sites are utilized. The basic responsibility for such programs on an item by item basis has been assigned to the cognizant AMC command. The Commanding General, USATECOM, however, is responsible for assuring existence of adequate surveillance test programs.

8-2.4 SEQUENCE OF TESTS

Fig. 8-1 illustrates a logical sequence of tests performed during the life cycle of any Army amphibious vehicle.

8-3 RESPONSIBILITY FOR VEHICLE TESTS

The scope of this discussion on test responsibility is limited to the specific areas of responsibility for prescribing the tests and associated test formats, and the responsibility for conducting the tests.

As indicated earlier in this chapter, the total

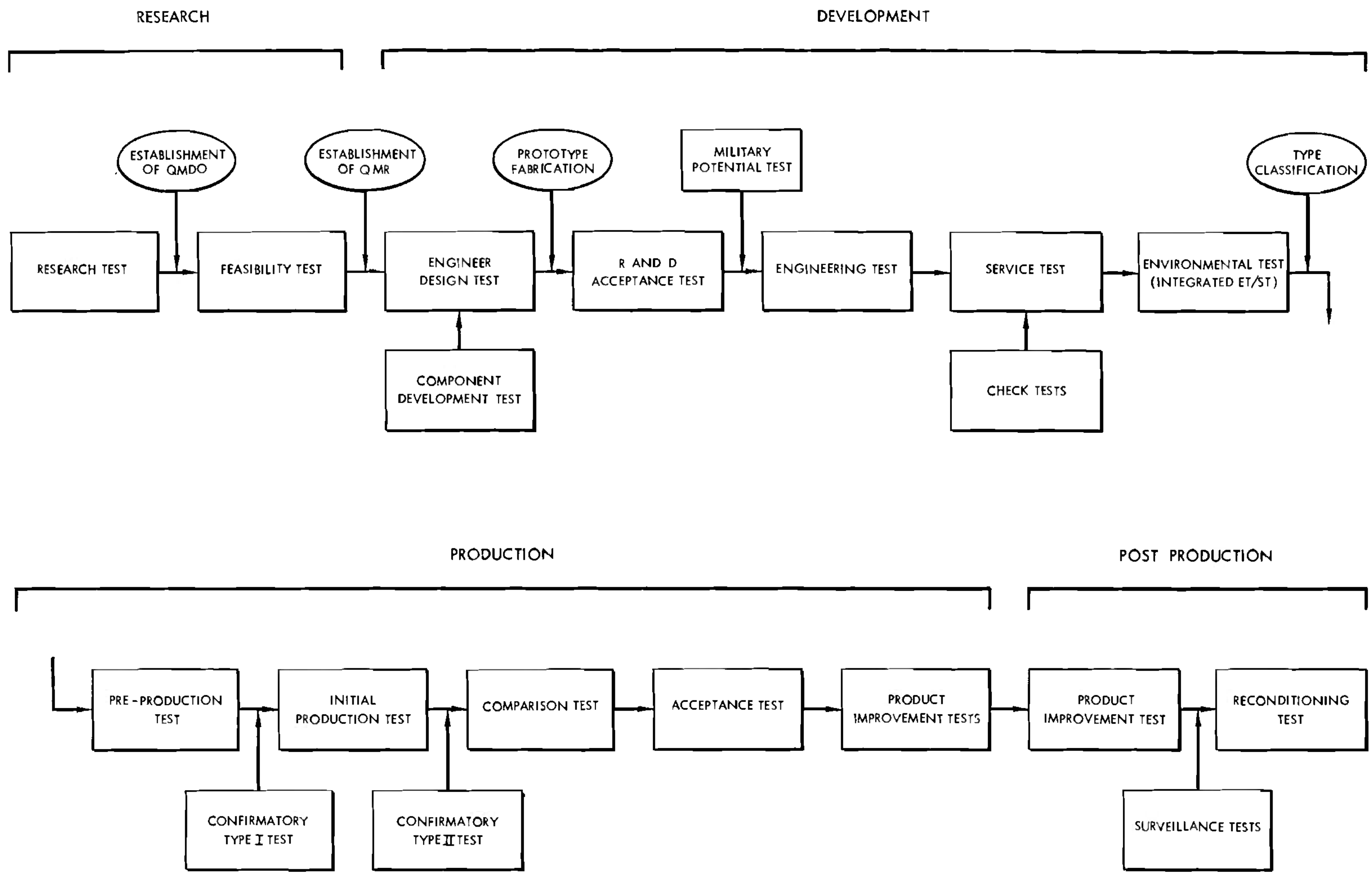


Figure 8-1. Testing Performed at Various Stages in the Life Cycle of an Army Wheeled Amphibian

responsibility for testing a vehicle with regard to Type Classification and Army acceptance falls under the USATECOM. Therefore, the responsibility of USATECOM is to insure that a vehicle meets the military requirements of the QMR. Results of these tests will be quantitative only insofar as the QMR or its interpretation is quantitative, i.e., Category I Test results will be of little or no use to the designer with regard to solving design deficiencies; these results will only indicate the problem areas.

It is the responsibility of the amphibian development command(s) to supply a vehicle to USATECOM which meets the military requirements and/or provide a "fix" when a performance requirement is not met. It logically follows (and is in fact the case) that responsibility for obtaining quantitative test data (design data) with which to predict vehicle performance and to correct design deficiencies lies with the cognizant development commands. The commands and agencies most likely involved with wheeled logistic amphibians, along with their areas of cognizance, are listed in Table 8-1.

The responsibility for administering a test program, conducting tests, and test data retrieval can be delegated by the cognizant development command(s) to private contractors, and/or USATECOM. Of course, the ultimate responsibility for the amphibian meeting the QMR and for rectification of design deficiencies lies with the amphibian project office, usually under the Mobility Equipment Command (MECOM).

In this regard, it is the responsibility of the project office to determine and to be aware of high risk areas evolving from the military requirements which may require test and evaluation during the amphibian development stage. For example, if requirements dictate a propulsion system or hull form which is unique or "border line state-of-the-art", a component development program including engine-transmission test or hull testing may be required. More specifically, in the case of engines and transmission systems, for example, test facilities at ATAC permit simulation of loads and environmental conditions on propulsion systems which exactly duplicate those experienced under any selected operational condition. It is under controlled test conditions such as these that high risk areas should be initially evaluated for several reasons—detailed and accurate engineering data are easier to obtain than on a completed amphibian, and it is obviously much more desirable to make as many design changes as possible before a prototype amphibian is built.

There are, of course, a number of trade-offs with regard to testing components before or after installation in the amphibian. These trade-offs involve such things as probability of failure (risk), scheduling, funding, and the availability of equipment and test site. Clearly, the responsibility for making the engineering

TABLE 8-1 AMPHIBIAN DEVELOPMENT COMMANDS

Command or Agency	*Cognizance*
U. S. Army Tank-Automotive Command (ATAC) Warren, Michigan	Engines, power trains, suspensions, tires, soil analysis, environmental factors.
U. S. Army Mobility Equipment Command (MECOM) St. Louis, Missouri	Amphibious Lighters
U. S. Army Mobility Equipment Research and Development Center (MERDC) Ft. Belvoir, Virginia	Research, development, testing, and evaluation of mobile equipment and amphibious vehicles.
U. S. Army Combat Developments Command Transportation Agency (CDCTA) Fort Eustis, Virginia	Development of Logistic Over-the-shore (LOTS) capability.

decisions, with regard to establishing an optimum test program for development, rests with the development command(s), the individual designers, and program officers.

With the above testing responsibilities defined, it is important that the designer be aware of how USATECOM achieves its prime responsibility—testing and type classification of materiel. The ultimate goal (Standard Type Classification) must be considered throughout the amphibian development cycle.

For the past several years, USATECOM has been engaged in a program of standardizing their test procedures to the greatest extent possible. This program entails the establishment of Materiel Test Procedures (MTP's) which outline, as specifically as possible, the tests performed on all possible items of Army materiel. At the present time, it is anticipated that the basic series of MTP's will be completed in the early 1970's; however, continual updating will be required.

Theoretically, at some future date, an entire test format will be developed by calling for the appropriate MTP's. The MTP's will provide the basic guidelines for amphibian testing; but the details and ramifications will be described within the framework of the set procedures or will supersede the MTP's, if necessary.

The significance of the MTP's to the amphibian designer is that an awareness of the tests, test objectives, and methods which will be applied to the amphibian, will enable him to better design an acceptable item. Also, it should give some insight as to what areas of design require quantitative test data retrieval or updating, such that the risk of test failure is minimized. It is possible that the test procedures should be updated to result in more quantitative results which will allow the designer to compare items on an equal basis, and evaluate design concepts and changes.

Materiel Test Procedures and information regarding the status of the above mentioned program can be obtained from the USATECOM, Aberdeen Proving Ground, Maryland. The basic concept of MTP usage and a comprehensive list of available and planned MTP's are included in Ref. 1.

In areas of low risk, it may be possible to obtain the necessary design test data at the same time USATECOM is conducting the Category I Tests, provided the tests are compatible and agreement can be reached between USATECOM and the cognizant development or commodity command.

There are several test sites available to the designer to which the responsibility for conducting amphibian tests may be delegated. Each of the various test sites has the environmental facilities for conducting tests of a specific type. For example, Aberdeen Proving Ground, Maryland, has extensive facilities for determining land mobility. Table 8-2 includes a list of the major amphibian test sites with their corresponding areas of test capability.

Other contractor test facilities are available to the designer through the amphibian manufacturer. The selection of appropriate test sites for an amphibian test program is obviously dependent on availability, logistics, and test requirements.

8-4 DESCRIPTION OF AMPHIBIAN TESTS AND TEST PROCEDURES

8-4.1 GENERAL

The detailed description of all possible amphibian tests including procedure, data accuracy, and instrumentation would take undue space. The details of test procedures are not as important to the designer as the desired end product or the basic concept motivating the test. In line with the scope of this handbook, an attempt will be made in the discussion which follows to indicate types of tests, the specific objectives of each type of test, the general methods or procedures involved, accuracy required, and application of test results by the designer.

Another limitation must also be imposed on the discussion of tests, i.e., quantitative or engineering tests will be covered in some degree of detail, and qualitative (e.g., endurance, service) tests will be mentioned only briefly. The reason for this is that qualitative test data are of value in indicating problem areas; however, where the solutions to these problems are not intuitive or obvious, the application of quantitative test data is generally the best means of reaching a sophisticated solution to a given problem.

A statement about accuracy is considered necessary at this time since the difference between qualitative and quantitative tests is

TABLE 8-2 AMPHIBIAN TEST SITES AND CAPABILITIES

Site	*Capabilities (Facilities)*
Aberdeen Proving Ground (APG), Maryland	Land mobility testing
Fort Story, Virginia	Water operations, LOTS, evaluation, land mobility, surf performance (limited)
Monterey, California	Surf tests
Yuma Proving Ground Yuma, Arizona	Environmental (desert), land mobility tests
Camp Pendleton, California	Land endurance testing, beach and waterborne performance testing
General Equipment Test Activity Fort Lee, Virginia	Water performance, beach operations and coordinated ship and amphibian operations
Harts Lake Proving Ground, Michigan	Land mobility and water testing
Warren Dunes, Michigan	Sand beach and dunes, water tests
Twenty Nine Palms, California	Desert environment tests

really an undefined difference in accuracy of test measurements, whether "go-no go", gross human observation and comment, or instrument readouts. In general, the degree of accuracy of test measurements is basically a function of such items as the test objective, the available instrumentation, sensitivity of the amphibian design performance to the measured parameter, cost, and time. As far as the designer is concerned, the motivation for greater accuracy at additional cost and time, is a direct function of the sensitivity or effect of the test data on the design and performance of the amphibian. As an example, the developed engine power varies approximately as the waterborne speed cubed (i.e., $HP \propto V^3$); therefore, in order to have a balanced effect, design water speed measurements might warrant more attention to accuracy than does power measurement. This example is presumptuous in that other factors such as guaranteed (under contract) power availability may dictate a high degree of accuracy in engine power measurements.

It should be noted that more accurate test data can give rise to improved methods of performance prediction, resulting in lower design margins or ignorance factors and, thus, enhance the optimization of amphibians by the evolution process.

8-4.2 ENGINEERING (QUANTITATIVE TESTS)

For convenience, the testing of an amphibian is divided into four categories.

1. Static tests, instrumentation, and calibration
2. Land tests
3. Water tests
4. Surf tests

The nature of an amphibian does not readily permit such a division or categorizing of tests since many functions and tests will overlap into other areas; however, the division is necessary for establishing a framework within which to discuss amphibian tests.

Instrumentation required for obtaining the desired test data will generally include thermocouples, pressure sensors, pyrometers,

gages, flow meters, load cells, scales, strip recorders, surveying instruments, stop watches, ohmmeters, thermometers, torsion meters, inclinometers, accelerometers, thrust meters, etc. These types of instruments are commercially available with accuracy ranges of 1 to 3 percent (plus or minus).

All instrumentation should be calibrated, both prior to and at completion of engineering tests. Calibration checks should also be made during tests if erratic readings are noticed or instrumentation damage is suspected for any reason.

In the test descriptions which follow, the types of instruments will be related to specific tests, and the significance of the data obtained will be noted where appropriate. In many tests, the usefulness of the test and instrumentation required is obvious. Refs. 2-4 provide additional information.

8-4.2.1 Static Tests, Instrumentation, and Calibration

The preparatory tests or checkout tests are grouped under this category, namely:

a. Measure and record physical dimensions and characteristics of amphibian.

b. Fill all fluid systems with prescribed type and amount of fluids (except fuel). Flush and bleed as required.

c. Check visually all lubrication points for lubricant and accessibility.

d. Check resistance of all electrical circuits with ohmmeter for open and closed resistance within required limits. Check insulation for adequate resistance.

e. Hydrostatic test of hydraulic and other pressurized fluid systems to 1.5 to 2.0 times working pressure and check for leaks.

f. Fill fuel systems and check for leaks.

g. Test operation of amphibian components and systems including:

(1) Starter system
(2) Engines
(3) Generator (alternator)
(4) Hydraulic pumps
(5) Steering mechanism(s)
(6) Transmission-drive train
(7) Brakes including emergency brake
(8) Lights
(9) Tire pressure and pressure control system operation
(10) Operation of all installed gages
(11) Winch and ramp operation
(12) Heaters and ventilators operation
(13) Cooling fans operation
(14) Bilge pumps operation
(15) Windshield wipers operation
(16) Exhaust system integrity
(17) Short test of power to marine propulsor
(18) Safety equipment, fire extinguishers, tools, etc.
(19) Check lifting rings, hand rails, towing eyes, hold down attachment points, etc.
(20) Check communication system operation
(21) Test hatches, cab, etc., for watertightness

The purpose of the static tests outlined is to insure that the amphibian is ready for the actual engineering tests to follow. The values of hydrostatic test pressure, electrical resistances, steer time, etc., should be obtained and checked against the amphibian specifications. All deficiencies and malfunctions should be corrected prior to conduct of further tests. Any corrective measures can usually be carried out during the instrumentation installation phase of the test agenda.

8-4.2.2 Land Tests

8-4.2.2.1 Determination of Weight, Center of Gravity, and Adequacy of Lifting Eyes

These three tests lend themselves to simultaneous performance in most cases. The object of the test is evident by its title. The reason for measuring weight and CG location is their relation to design weight and amphibian stability, respectively.

The first step of the test procedure is to accurately record the condition of the amphibian with regard to fuel load and location of movable items of onboard equipment.

Weight of the amphibian is usually determined by measuring and recording weight on each wheel. The center of gravity location is located by hanging the amphibian from slings as shown in Fig. 8-2. A vertical line from the point of suspension will always pass through the *CG*. By heeling the amphibian in both directions, the vertical and transverse location of the *CG* is indicated by the point at which these (plumb) lines do intersect.

NOTE: Both of the sling methods indicated in Fig. 8-2 locate the vertical center of gravity. This provides a cross check of the tests.

By recording all added or subtracted weights and their location, the weight and *CG* location of the amphibian can be determined for any given condition.

8-4.2.2.2 Maximum Draw Bar Pull and Towing Pad Tests

Fig. 8-3 shows a method for conducting towing pad tests. A specified load is applied, and after the test the tow pads and attachment structure are checked for evidence of failure or overstress. The tension is applied slowly in increments. Then, if a failure does occur, a reasonable estimate of the failure load may be made. Measurements of pad eye diameter and length are made prior to and following the test in order to detect any deformation.

Maximum draw bar pull tests are conducted in much the same manner as the towing pad tests. However, only one end of the amphibian is bridled and the tension is applied by the engines through the tires of the amphibian. The engine speed is set at or near the maximum torque rpm with torque converter in a stall condition (static vehicle) and the maximum attainable tension is measured. Temperatures and pressures of engine oil, cooling water, torque converter fluid, etc., must be closely monitored during this test so as not to cause failure or damage.

During this test, torsion meters and tachometers may be employed to determine engine power and torque converter characteristics.

8-4.2.2.3 Bilge Pump Tests

Bilge pumps are tested in order to determine that capacity requirements are met. For each pump, the discharge capacity is measured as a function of the prime mover rpm. This is usually accomplished by maintaining a level of water in the amphibian's bilges and recording the time required for the pump to discharge a predetermined quantity of water. Water flow meters may also be used for this test.

By recording elapsed time and measured water quantities (or flow rates) for increments of prime mover speed (rpm), the bilge pump capacity (gpm) can be determined as a function of pump or engine speed. It should be noted that bilge pumps are usually driven hydraulically by the main engine.

Another measurement which is recorded is the elapsed time between actuation of bilge pumps and start of discharge.

Hand or emergency bilge pumps are similarly tested.

8-4.2.2.4 Minimum Turn (Steering) Radius and Response Time Tests

The amphibian is driven over a surface which will clearly define the tire tracks of all wheels. Condition of the amphibian (i.e., weight, tire pressures, etc.) is recorded (or specified). For "hard over" turns to the right and left, for both two-wheel and four-wheel steering, if applicable, the mean radius of the center of the inside wheel track and outside wheel tracks for a 360 deg turn, are measured and recorded. See Fig. 8-4.

The steering response test measures the elapsed time for steering from stop to stop or hard right to hard left. The time varies with amphibian load, tire pressure, engine speed (hydraulic pressure and flow), and surface conditions.

8-4.2.2.5 Winch Tests

For various engine speeds (hydraulic pressures and flow rates), line speed is measured in feet per second for pulling a known heavy load mounted on wooden skids over dry pavement. From these test data, winch capability can be established and compared with design data.

8-4.2.2.6 Tire Inflation System Tests

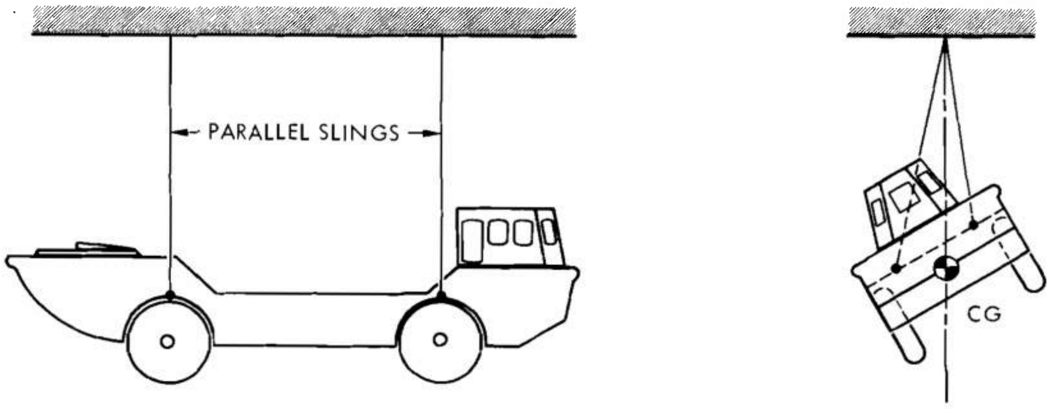
Each tire is inflated to an overdesign pressure and isolated from the compressor system. Pressure loss is recorded after 24 hr.

8-4.2.2.7 Ramp Tests

The ramp is repeatedly opened and closed. The time for each operation is measured and related to various parameters such as engine speed, number of engines running, and hydraulic pressure and flow rate.

8-4.2.2.8 Fan (Cooling) Tests

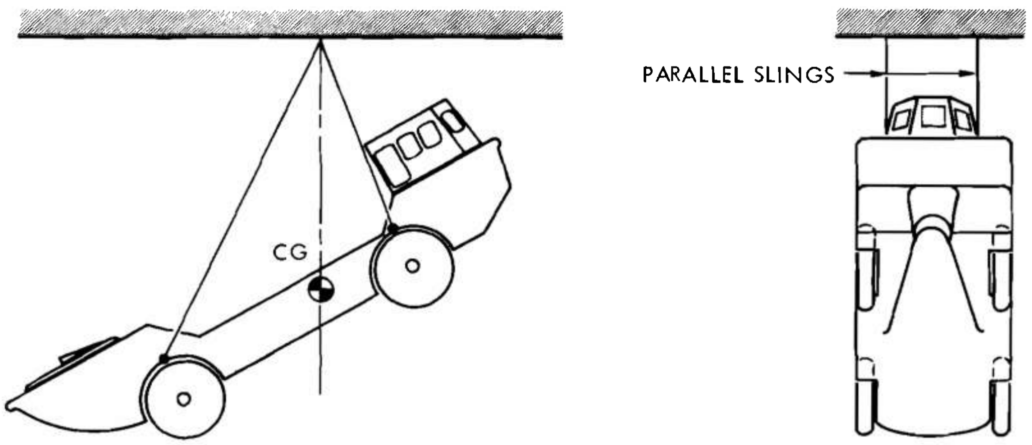
Each fan is tested over a range of engine rpm



MAINTAIN ZERO INCLINATION DURING HEELING OF LIGHTER ATHWARTSHIPS

HEEL AS SHOWN ABOVE (30° min) REPEAT FOR HEEL OF OPPOSITE HAND

(A) TRANSVERSE AND VERTICAL CENTER OF GRAVITY DETERMINATION



INCLINE AS SHOWN ABOVE (30° min) REPEAT FOR INCLINATION OF OPPOSITE HAND

MAINTAIN ZERO HEEL DURING INCLINING OF LIGHTER FORE AND AFT

(B) LONGITUDINAL AND VERTICAL CENTER OF GRAVITY DETERMINATION

Figure 8-2. Center of Gravity Determination

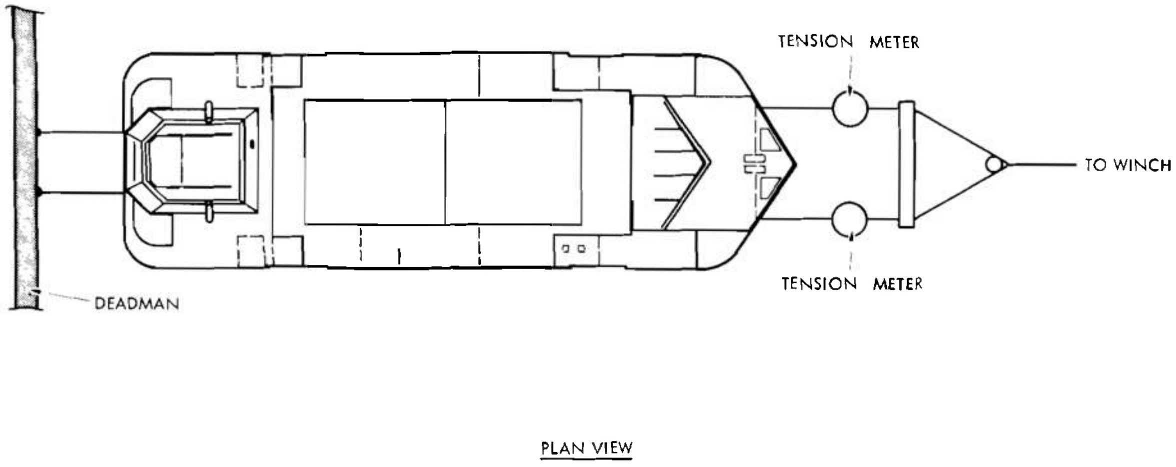


Figure 8-3. Towing Pad Test Procedure

and fan rpm. Air temperatures (to and from the fan), velocities, and flow rates are measured. These data are used to determine the fan's cooling capacity.

8-4.2.2.9 Brake System Tests

With the amphibian in a known load condition, emergency stops are made on dry pavement from a range of speeds including reverse. The speeds and corresponding stopping times, and distances are measured and recorded. Accelerometers are sometimes used in these tests to record braking "g" levels.

8-4.2.2.10 Parking Brake Tests

Two types of tests are performed on the system. One is a static pressure test which measures loss of pressure over a period of hours. Another is determination of the maximum grade on which the brake will hold the loaded amphibian in both forward and reverse directions at the design holding pressure.

8-4.2.2.11 Speed Trials (Maximum)

Speed trials are conducted on a level, dry, improved surface. At a specified load condition, the amphibian is run through a measured distance at maximum (constant) engine speed in forward high gear and the elapsed time determined. Velocity is then calculated. Runs are made in both directions and sufficient distance is provided for acceleration to constant maximum speed prior to entering the measured course.

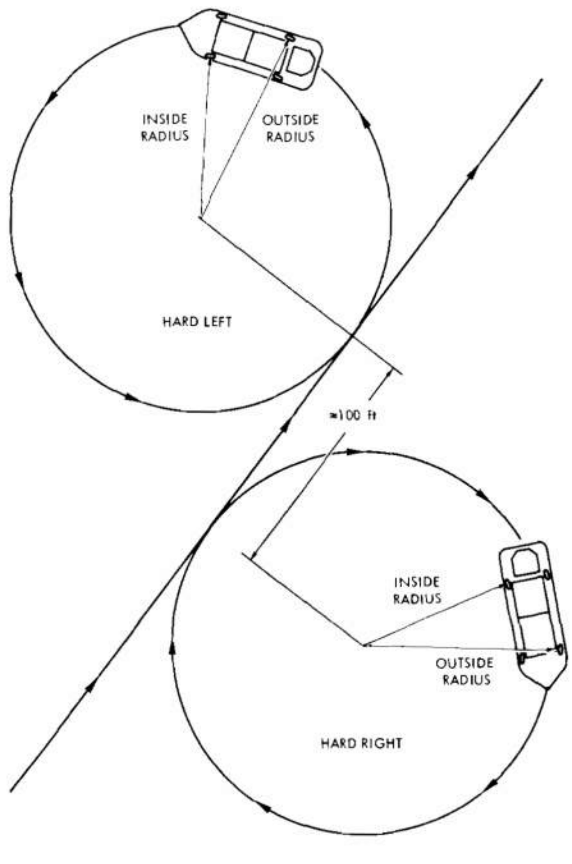


Figure 8-4. Diagram of Steering Radius Test

8-4.2.2.12 Fuel Consumption Tests

The fuel consumption (as a function of speed) tests are usually run for full and light load conditions on a level improved surface. The method used to measure fuel consumption is to time the consumption of a premeasured quantity of fuel which is placed in a calibrated auxiliary fuel tank. This tank can be cut in and out of the fuel service system as required.

8-4.2.2.13 Gradability Tests

With the amphibian in low gear, it is slowly driven up a gradually increasing grade until forward motion is impossible due to the grade. The test conditions which may be varied include grade surface condition, tire pressure(s), and load condition. Care should be taken not to over heat, over torque, or cause damage to the power system.

Results of this type of test are sometimes presented in terms of maximum attainable speed for various specified grades. Again, these data are functions of road surface, load, etc.

8-4.2.2.14 Rolling Resistance Tests

The purpose of this test is to determine the rolling resistance of the amphibian as a function of speed for various land surfaces.

One method of obtaining this information is to run the amphibian through a calibrated (posts or stripes placed at regular intervals) course at constant speed. Upon entering the course, power is removed from the wheels (declutch) and the amphibian is allowed to coast to a stop. Movie films are taken and analyzed to determine the change in amphibian speed versus time. From these data, the deceleration force (rolling resistance) can be calculated.

The same method can be used in braking tests to determine braking force, i.e., total deceleration force = rolling resistance + braking force.

Another method which can be used to determine the rolling resistance is to tow the amphibian at constant speeds and measure the towing forces with a tension dynamometer.

8-4.2.2.15 Off-road Mobility and Obstacle Tests

The amphibian is run over prescribed courses consisting of swamps, marshes, mud slopes, clays, and sands to determine its off-road mobility—i.e., the range of soil conditions over which it can operate—and to define the operating limitations. Similar tests are conducted to determine the ability and limitations (go or no-go) of the amphibian to negotiate specified obstacles such as vertical banks and ditches.

A related test is used to determine the amphibian's bank egress capability. However, the conditions are very difficult to define and measure for this particular type of test.

8-4.2.3 Water Tests

8-4.2.3.1 Stability Tests

The stability tests are performed to determine an amphibian's righting moment (i.e., stability) for roll angles up to the point where capsizing is imminent. Tests are run for both light and loaded conditions.

Fig. 8-5 shows a typical method of inclining the amphibian with a series of cable attachments, winches, and anchors. This figure also indicates the data recorded (i.e., cable tension, roll angle, and rigging geometry) along with the method of calculating the righting moment. It is important that the amphibian weight and center of gravity be known by relating added weights and their locations back to the light condition center of gravity determined during land tests.

Heel angles should be increased by approximately 5 deg increments until the heeling force (cable tension) drops off sharply. This drop in tension indicates approaching instability (imminent capsizing) and the test is then discontinued. Safety ropes or cables must be used during these tests in order to prevent possible capsizing, but these ropes must not restrain the heeling.

The data obtained provide a good measure of waterborne static stability and can easily be compared to other amphibian data. An accepted means of representing these data and typical test results is shown in Fig. 8-6.

8-4.2.3.2 Moment to Trim One Inch (MT1)

For a specified loading condition, the moment required to cause the sum of forward

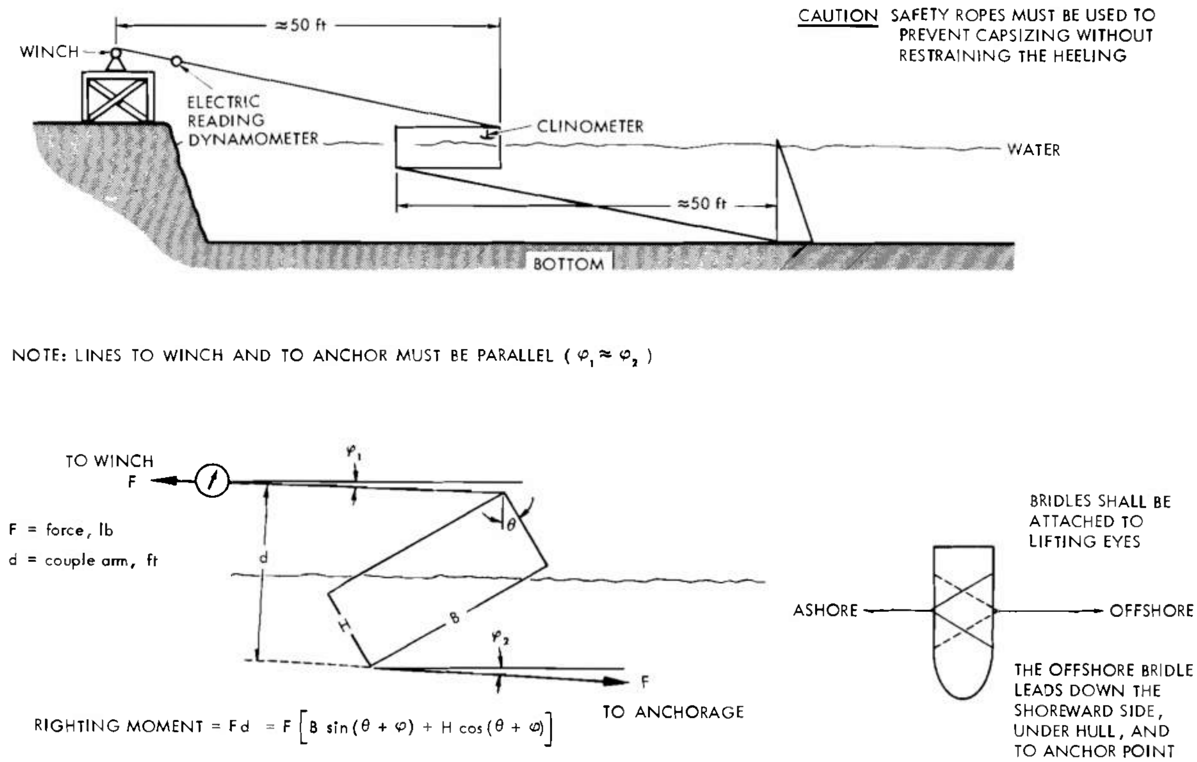


Figure 8-5. Righting Moment Determination by Inclining Test

waterline elevation plus the aft waterline depression to equal one inch is determined. The locations (stations) at which waterline measurements are made are usually specified and/or coincide with the fore and aft draft markings.

The moment variation is accomplished by moving a weight longitudinally along the amphibian centerline.

Data obtained from this test confirm the design objectives and are used to determine the best cargo placement in order to minimize waterborne resistance or to maximize directional stability since these characteristics are a function of trim.

8-4.2.3.3 Speed Trials (Power and Fuel Consumption)

Waterborne speed trials are accomplished in much the same manner as land speed tests, except that the added complications of determining the exact position of the amphibian and straight course running are involved. Fig. 8-7 shows a typical water course from which the methods used for orientation and course keeping may be inferred.

The amphibian is run through the calm water course at specified (constant) engine speeds, and elapsed time is measured. Waterborne speed is then calculated.

Power, trim, fuel consumption, engine data, etc., may also be obtained at this time, if required.

Power measurements are made in several ways depending on the accuracy required. The easiest, and usually least accurate method, is to record engine speed and fuel rack (throttle) setting. With these data and the manufacturer's engine curve, an estimate of power *(BHP)* can be made. Power can also be calculated from measured torque (torsion-meter) and rpm, or from propeller thrust(s) and amphibian speed.

8-4.2.3.4 Tactical Diameter (Turning Circle) Tests

This test determines the waterborne port and starboard turning circle diameters for various conditions as specified or required. Many variables influencing the test results must be

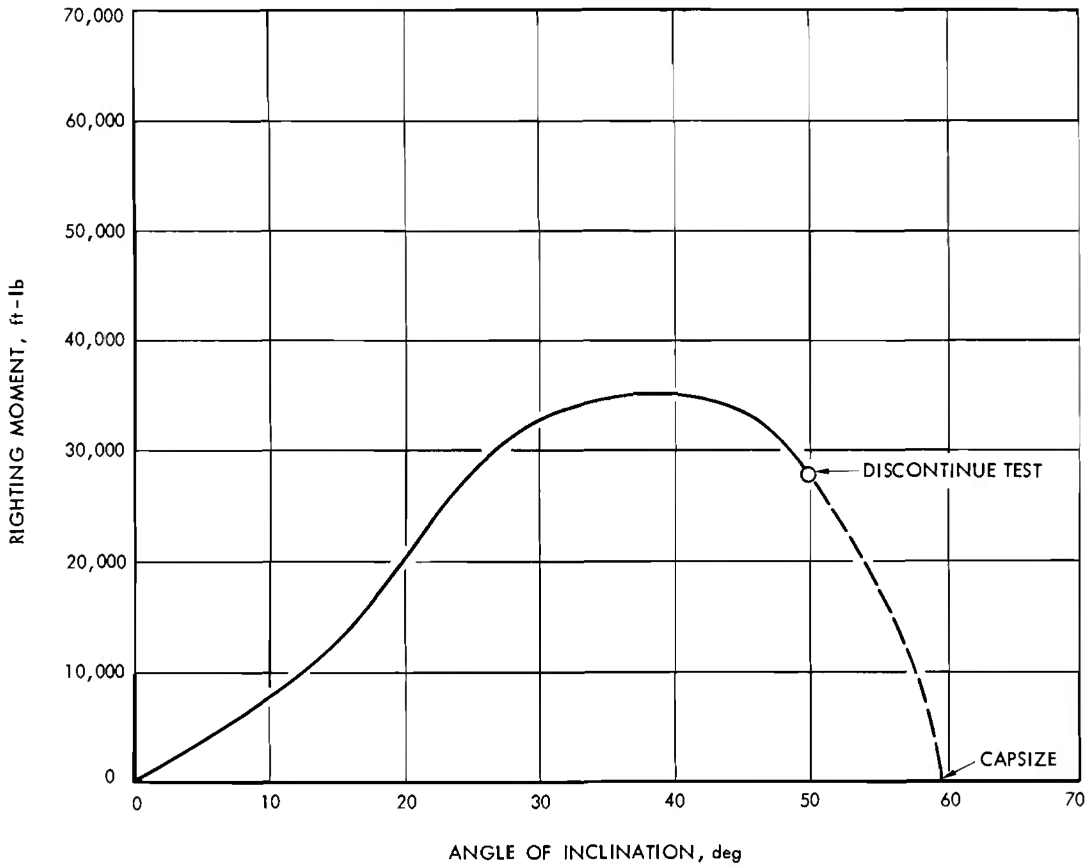


Figure 8-6. Typical Waterborne Statical Stability Curve for an Amphibian

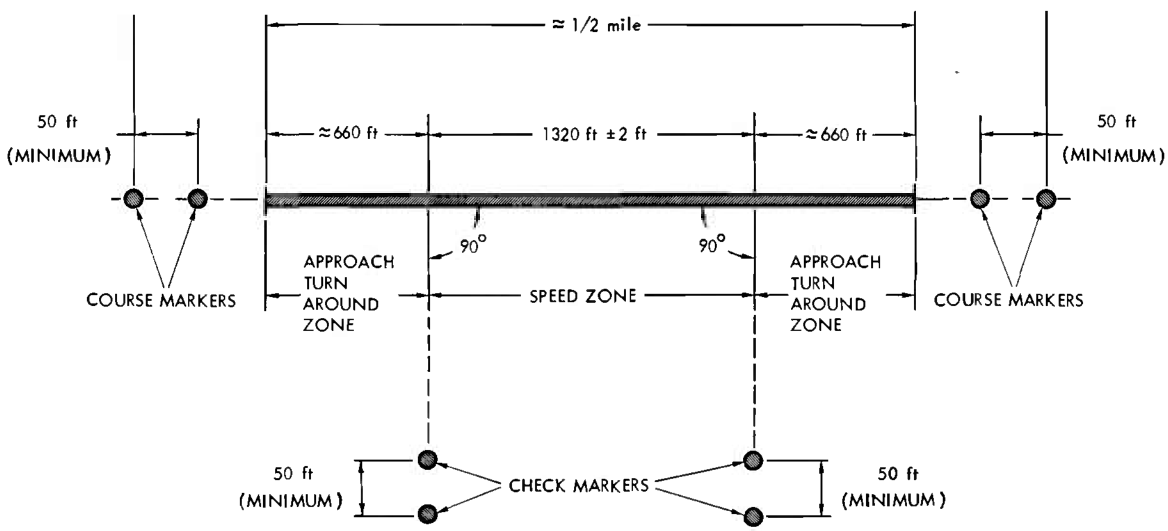


Figure 8-7. Plan View of Typical Water Course for Amphibian Speed Runs

taken into consideration including load condition, approach speed, and rudder angle set at execution of turn. If two propulsion devices are employed, the minimum turning circle will result when reverse or astern power is applied to the inboard (with respect to direction of the turn) propulsor.

As with the waterborne speed trails, the determination of the amphibian's position and path during the turn is the complication. Methods which have been used in the past include calculating the amphibian's position by triangulation from shore stations at regular intervals during the execution of a turn; see Fig. 8-8. Another method consists of dropping buoys at regular intervals during the turn and measuring the turning circle diameter by triangulation or direct tape measurement.

8-4.2.3.5 Crash Stop Tests

These tests are used to determine minimum distance (ahead reach) from the point of execution of astern power to a dead stop in the water.

The method used is to approach a marked course at full speed. When range posts are lined up, astern power is applied to the propulsor. Distance is then measured or triangulated from point of execution to the point where forward motion is stopped.

During this maneuver, care should be taken not to over torque the engines; close monitoring of the machinery operation should be maintained.

8-4.2.3.6 Astern Operation and Maneuvering Tests

The amphibian is operated with full power astern for a sustained period of time. During this period, rudder response is recorded and the sustained astern power capability is evaluated in terms of pressures, temperatures, vibration, etc. One method of evaluating rudder response is to measure the rate of change in heading (compass) at maximum astern speed for various rudder

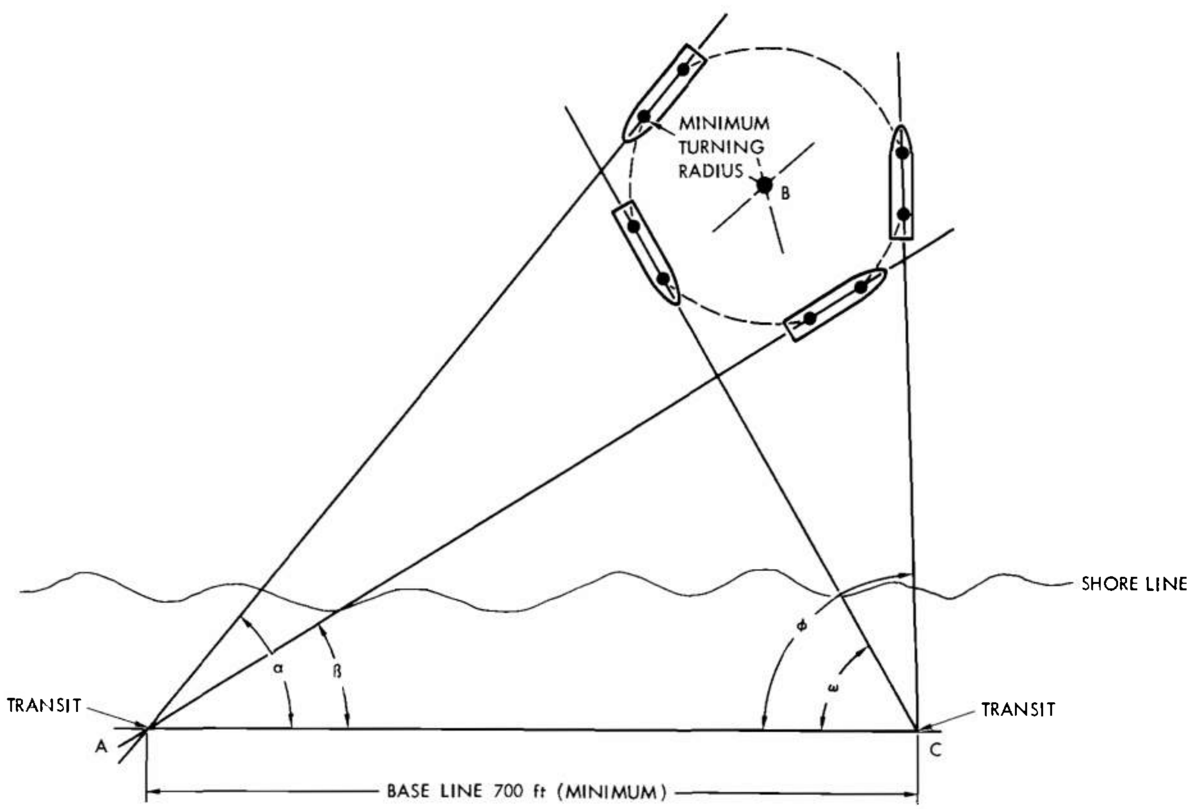


Figure 8-8. Method for Determination of Waterborne Turning Radius

angles. Rudder angle required to maintain a steady (straight) course astern should also be recorded.

8-4.2.3.7 Bollard Pull Tests

Bollard pull tests are somewhat analogous to the draw bar pull tests on land. The amphibian is moored to a rigid structure such that the line lies in the vertical plane passing through the propeller shaft or line of thrust. The mooring line should also be as close as possible to horizontal.

A tension dynamometer is placed in this line, and tensions recorded for increments of propeller or propulsor speed (rpm).

For multiple propulsor amphibians, each system should be tested independently. See Fig. 8-9 for a sample test method.

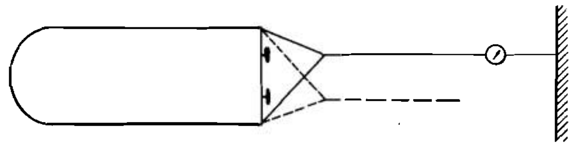


Figure 8-9. Bollard Pull Test Method

8-4.2.3.8 Towing Tests

These tests are performed to determine an amphibian's towing capability and its ability to be towed. Selected vehicles are towed by the test amphibian and tow line tension, speed, and power are measured.

The test amphibian may also be towed in various load conditions without propellers to determine its waterborne resistance at various speeds of advance. The effective horsepower *(EHP)* can then be calculated.

8-4.2.4 Surf Tests

The state-of-the-art in surf testing amphibians with regard to qualitative analyses is very limited. One of the major variables involved in surf testing is the capability of the operator. With this in mind, the test methods which follow represent those that attempt to determine surf performance.

Under various load conditions and for various surf heights, the amphibian is run through the breakers in both directions, i.e., landing on the beach and re-entering the water. Films are taken of the tests and degree of directional stability is measured in terms of the amphibian's extent of "wallowing", tendency to broach, and amount of rudder control required. Another important factor with regard to surfing ability is the ability of the amphibian to withstand water impact of breaking waves over its cab and decks, and the ease with which this water is shed.

Surf tests in the past have simply compared a new design amphibian with other amphibians by operating them side by side in the surf. Evaluation is made by the operators, observers, and, at times, by the surf (when sinking results).

8-4.2.5 Other Tests

Other types of tests which are performed on amphibians during typical confirmatory testing, are tabulated in Table 8-3. The breakdown is, for the most part, self-explanatory. It is recommended that test reports for previous amphibians be reviewed for greater detail with regard to these tests.

Endurance testing involves much the same procedure as the confirmatory type test. However, many operational hours are accumulated and gross amphibian reliability, maintainability, and logistic requirements may be determined.

During the design phase of an amphibian, the designer should be aware of all the standard tests, and be acutely aware of the possibility of improving these tests and developing new tests as required for the particular amphibian with its related areas of high risk, high performance sensitivity, and degree of complexity.

TABLE 8-3 TYPICAL TYPE II CONFIRMATORY TEST AGENDA

A.	Familiarization and Training
B.	Preoperational Inspection
C.	Road Mobility
D.	Off-road Mobility
E.	Fuel and Oil Consumption
F.	Ease of Operation (Crew Safety and Comfort)
G.	Maintenance Evaluation
H.	Water Operations
I.	Towing Test
J.	Surf Performance
K.	Operation with Landing Ships
L.	Loading and Unloading
M.	Movement Adaptability (Transportability)

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2. *Standard Tests for Engineering Testing of Amphibious Landing Vehicles (Tracked)*, Report No. 105-1-1004, Selwood Research, Inc., Kensington, Md., June 1967.
3. *Ordnance Proof Manual*, Vol. II, *Automotive Testing*, Aberdeen Proving Ground, Maryland.
4. *Surf Testing Procedure*, Amphibian Vehicle Division, Landing Forces Development Center, Quantico, Virginia.

PART FOUR HIGH SPEED AMPHIBIANS

CHAPTER 9 HIGH SPEED AMPHIBIANS

9-1 INTRODUCTION

There has been considerable interest in the design of wheeled amphibian vehicles with high water speed since the late 1950's. The first studies of high-speed amphibians were initiated by U. S. Army Transportation Evaluation Command, Fort Eustis, Virginia, under contract with Ingersoll-Kalamazoo Div., Borg-Warner Corp., Kalamazoo, Michigan. After a design study, further work was suspended due to pressing work on the development of the LARC V and LARC XV, and limited funding.

A study conducted by the U. S. Army Transportation Combat Development Group (Ref. 1) indicated that a high water speed will greatly increase the productivity of a logistical amphibian if the one-way water distance is on the order of 10 miles or greater. Thus for some mission requirements a wheeled amphibian with a high water speed may be desirable.

During the period from the early 1960's until 1968, the Amphibian Vehicle Division, Development Center, Marine Corp Development and Education Command, conducted an extensive study of high-speed wheeled amphibians. In order to increase the water speed, the study revealed that—among other things—the hydrodynamic drag must be reduced below that of the conventional wheeled amphibian displacement hull. This fact led naturally to the consideration of other hull configurations including planing hulls, hydrofoils, surface effect or air cushion craft, and catamarans.

As may be expected, the waterborne performance requirements will have the greatest single influence on the characteristics of a high speed amphibian. Thus the performance required by the craft's mission must be carefully considered and defined. This is particularly true with respect to sustained speed in waves, range or endurance, and maneuverability. It should be noted that a high-speed amphibian may have its speed limited by the available power or its motions in waves. The power-limited speed is simply the highest speed possible in a given sea condition using all of the available power. The motion-limited speed is the speed at which the amphibian's wave induced motions, and in particular the acceleration, exceed the tolerance of the crew. The selection of these required speeds should be consistent with the other aspects of waterborne ability (i.e., loading at ship side and surfing) and the expected occurrence of various sea conditions. Ref. 2 provides some information on the latter topic. The price—in terms of weight, complexity, and cost for a high speed amphibian with performance in excess of that actually required for the mission—will be very high.

This chapter on high (water) speed amphibians is intended to provide data and projections of performance and characteristics of different vehicle types, and information on design problems that are unique to high speed amphibians. The design procedures and much of the data presented in previous chapters apply to high-speed amphibians and thus will not be repeated. No attempt will be made to present detailed procedures for the hydrodynamic design of a high-speed amphibian since this would require an effort beyond the scope of this handbook. However, a list of suitable references for each vehicle type is included.

9-2 PLANING HULL AMPHIBIANS

One possible type of high-speed amphibian is based on the use of a planing hull form. On a planing hull form a large part of the weight is carried by dynamic lift at high speed. There is little wavemaking drag at high speed so that the power required is reasonable. At intermediate speeds (15 to 20 mph for a 35-ft vehicle) the wavemaking drag is high which may, in fact, fix the power required. This problem is discussed in par. 9-3.1.

9-2.1 EXISTING PLANING HULL AMPHIBIANS (LVW-X1 and LVW-X2)

Numerous planing hull amphibian forms were studied and model tested by the Stevens Institute of Technology in the late 1950's and reported in Ref. 3. From this work evolved the concept that was ultimately built. In early 1960 a contract was let by the Navy Bureau of Ships to the Ingersoll-Kalamazoo Div. of the Borg-Warner Corp. for the design and fabrication of two planing hull amphibians designated LVW-X1 and LVW-X2. The two vehicles were accepted by the Marine Corps in early 1964. The support of these vehicles was later taken over by the Chrysler Corp.

The LVW-X1 and LVW-X2 are nearly identical vehicles. Both have identical all welded aluminum "V" bottom hulls, retractable propellers, retractable aft wheel flaps to increase the planing surface, and an unsprung suspension system with fully retractable wheels. The major differences are that the LVW-X1 employs side loading and 2-wheel, 4-wheel, or oblique steering; whereas the LVW-X2 employs stern loading and 2-wheel steering. The LVW-X1 is also equipped with small trim control wedges on the wheel flaps.

The LVW's are each powered by a Lycoming TF-2036 gas turbine which drives the propellers, the wheels, or both as required.

For waterborne operation, two propellers and rudders are provided. These are mounted on a hinged propeller retracting plate which provides a smooth flat bottom for high speed planing operation. This plate retracts by pivoting around its forward end. Thus in the retracted position the propellers are in a tunnel, protected for land and surf operations.

In the land mode of operation, four-wheel drive is provided. Wheel retraction and extension are handled by a pivoting mechanism which allows the wheel to swing in the fore and aft direction. The operator's cab is located forward of the cargo well and is designed for both land and water operation with a crew of two. The fully enclosed cab is fitted with 3/4-in. shatterproof glass for protection in surf operation.

Figs. 9-1 and 9-2 show an LVW operating at high speed in the water and on land, respectively. The vehicle characteristics of the LVW-X1 and LVW-X2 are given in Tables 9-1 and 9-2, respectively; and an outline drawing of the LVW-X1 is given in Fig. 9-3.

9-2.1.1 Waterborne Speed and Power of LVW-X1 and LVW-X2

As indicated in par. 9-1, it is necessary to carefully define the waterborne speed of a high-speed amphibian and the conditions under which it was obtained: In order to obtain these data, the U. S. Marine Corps conducted a series of engineering tests on the LVW vehicles. The results are summarized in Ref. 4. After initial testing it was realized that, because of problems with weight and various components, the LVW vehicles should not be considered as prototypes for a planing hull amphibian. As a result, they were tested in the water as test bed craft for planing hull amphibian performance. These tests were conducted at a gross weight of about 38,000 lb which corresponds to the original full load design displacement. Tests at the actual full load displacement of 44,000 lb would not have been meaningful because of the increased weight over the design conditions.

The measured thrust horsepower, shaft horsepower, and propeller rpm as a function of speed in calm water for the LVW-X1 are presented in Fig. 9-4. The lines represent model test data scaled to full size and the points represent measured full scale data. An important item to note in this figure is the peak or hump in *SHP* at a speed of 20 knots, i.e., it requires as much power to reach 20 knots as 30 knots. This hump in *SHP* is due to high drag from wavemaking, poor flow conditions to the propeller, and low propeller efficiency due to a high thrust coefficient (see par. 7-7). This condition is typical of high-speed amphibians and means that the power required must be checked over a range of speeds in addition to the design point.

The actual top speed obtained by the LVW vehicles in calm water and Sea States 1, 2, and 3 are presented in Table 9-3. The speeds given in Table 9-3 represent the speed obtained when operating at full power. The Sea State numbers given refer to Table 9-4.

9-2.1.2 Accelerations of the LVW Vehicles

The LVW-X2 vehicle was instrumented and the accelerations in head seas were measured at

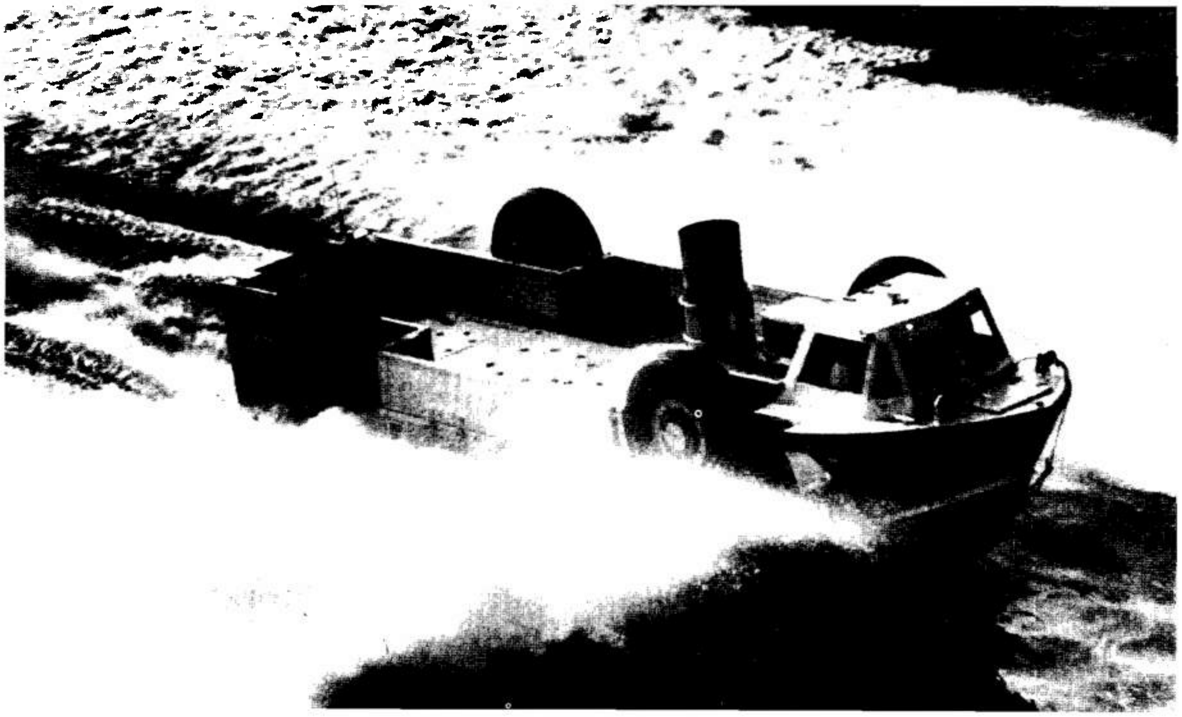


Figure 9-1. LVW-X1 in High-speed Waterborne Mode

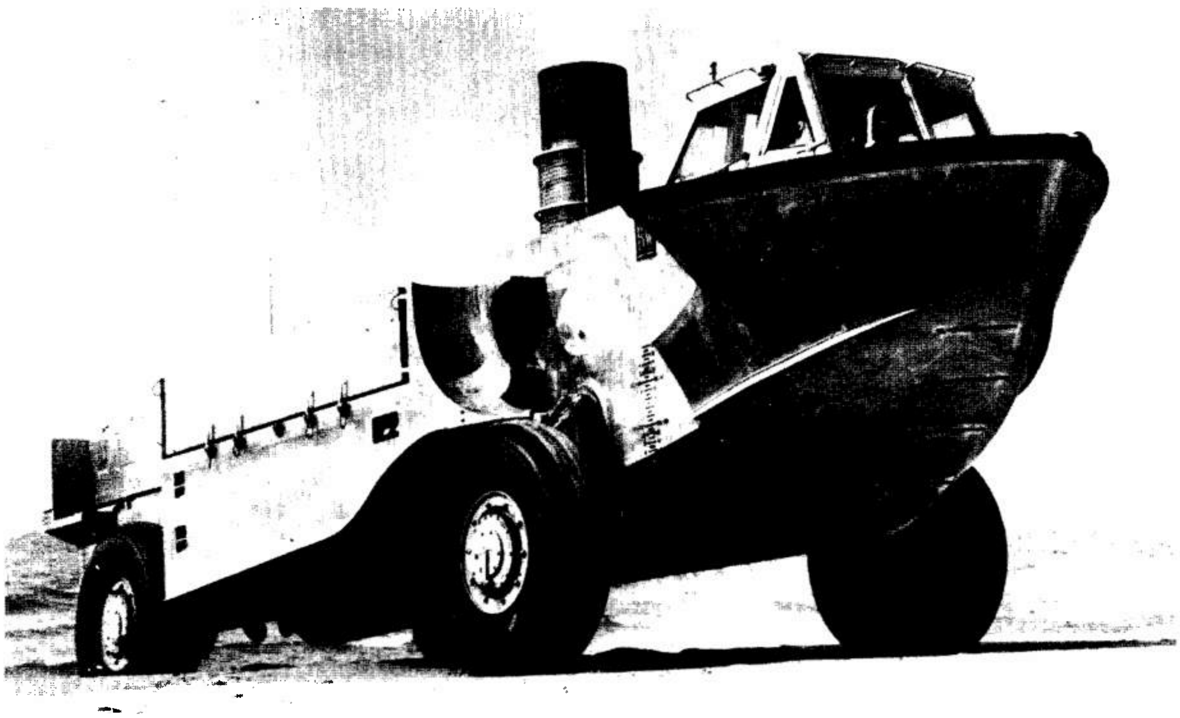


Figure 9-2. LVW-X2 in Land Mode

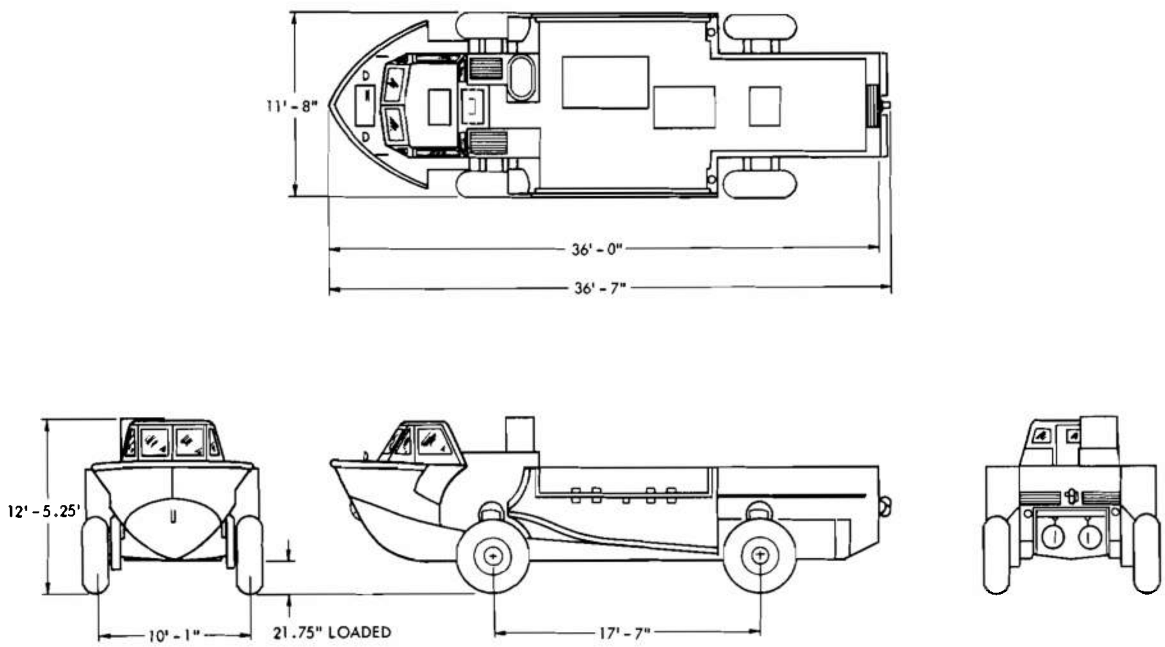


Figure 9-3. External Dimensions of LVW-X1

the bow (crew cab) and at the center of gravity. These data are presented in Ref. 4. Fig. 9-5 presents the average of the 1/3 highest upward (or impact) accelerations in "g's" of the LVW-X2 as a function of nondimensional wave heights and volume Froude numbers. The average of the 1/3 highest occurrences of a random variable such as wave height or vehicle accelerations is often called the significant value. The volume Froude number is a nondimensional measure of speed for a planing hull the same as the conventional Froude number or speed length ratio is for a displacement hull (see par. 7-4.1.1.3). The relationship between displacement, speed, and volume Froude number is given by:

$$F_{\nabla} = \frac{V}{\sqrt{g\left(\frac{\Delta}{\rho g}\right)^{1/3}}} \tag{9-1}$$

where

F_{∇} = volume Froude number, nd
V = speed, ft/sec
Δ = displacement, lb
g = acceleration due to gravity, ft/sec^2
ρ = mass density of water, slug/ft^3

A nomograph for the solution of Eq. 9-1 is presented in Fig. 9-6.

The data in Fig. 9-5 apply to the condition of head seas. In following seas, the accelerations are reduced considerably. Table 9-5 presents this reduction for typical planing hulls.

As previously indicated above, in order to define a "motion-limited speed" it is necessary to know the motions as a function of speed and sea state (wave height), and the tolerance level of the crew to the motions. During the testing of the high-speed amphibians some data on the crew tolerance were developed. These data are summarized in Table 9-6.

The reactions given in Table 9-6 are based on a limited sample and thus, must be regarded with caution. They are, however, the only data known to exist in which the crew reactions and measured accelerations were obtained simultaneously for high speed marine craft. More data of this type will be developed in the future, and as it becomes available should be used to revise Table 9-6.

If a waterborne transit time of 1 hr is assumed, the allowable significant upward acceleration level would be 0.9 g. In this case the actual sustained speed of the LVW in various sea states would be as given in Table 9-7.

TABLE 9-1. LVW-X1 VEHICLE CHARACTERISTICS

Manufacturer	Ingersoll-Kalamazoo Division of Borg-Warner
Width, ft-in.	11-8
Length, ft-in.	36-7
Height (Top of Cab), ft-in. (Top of Stack), ft-in.	11-0 12-5¼
Min Ground Clearance, in.	21¾
Wheelbase, ft-in.	17-7
Track Width, ft-in.	10-1
Angle of Approach, deg	25
Angle of Departure, deg	26
Break Angle, deg	15
Land Speed (Governed), mph	35
Water Speed	See par. 9-2.1.1
Land Range, mi	330 at 25 mph
Turn Radius (Land)	See par. 9-2.1.3
Turn Radius (Water)	See par. 9-2.1.3
Cargo Capacity (Design), lb	10,000
Cargo Space (L × W), ft-in.	21-10 × 10-10; Area = 183 ft^2
Total Weight, lb	44,000
Grade Ability (Design), %	60
Side Slope (Design), %	30
Surf Ability (Design), ft	10
Engine Data	Lycoming TF-2036-B1A Gas Turbine, 1,500 *BHP* (cont.) at 3,600 rpm and 80°F
Fuel Type	Marine Diesel, JP-3, -4, or -5
Fuel Capacity, gal	806
Marine Propulsor	(2) propellers; D = 24 in., P/D = 1.3, BAR = 0.945; Retractable
Suspension	Unsprung rigid system, 4 × 4
Brake System	Fawick, Caliper, Hydraulic-actuated, 4 sets of spot disc type
Tires	18 × 25 in., 12-ply
Land Steering	Hydraulic; Selective 2-wheel, 4-wheel, or oblique
Marine Steering	Hydraulic-actuated twin rudders
Electrical System	24-V DC, Single-wire
Hydraulic System	3,000 psi, 15 gpm at 2,000 rpm; 3,000 psi, 33 gpm at 2,400 rpm
Crew	2
Hull	Welded Aluminum 5456
Planing Surface	Retractable plate with mounted propeller

TABLE 9-2. LVW-X2 VEHICLE CHARACTERISTICS

Manufacturer	Ingersoll-Kalamazoo Division of Borg-Warner
Width, ft-in.	11-7¼
Length, ft-in.	36-7¼
Height (Top of Cab), ft-in. (Top of Stack) ft-in.	11¾ 12-5½
Min Ground Clearance, in.	21½
Wheelbase, ft-in.	17-7
Track Width, ft-in.	10-1
Angle of Approach, deg	25
Angle of Departure, deg	25
Break Angle, deg	15
Land Speed (Governed), mph	35
Water Speed	See par. 9-2.1.1
Land Range, mi	330 at 25 mph
Turn Radius (Land)	See par. 9-2.1.3
Turn Radius (Water	See par. 9-2.1.3
Cargo Capacity, lb	10,000
Cargo Space (L × W), ft-in.	21-10 × 10-10 ; Area = 180 ft^2
Total Weight, lb	44,000
Grade Ability (Design), %	60
Side Slope (Design), %	30
Surf Ability (Design), ft	10
Engine Data	Lycoming TF-2036-B1A Gas Turb., 1,500 *BHP* (cont.) at 3,600 rpm and 80°F
Fuel Type	Marine Diesel, JP-3, -4, or -5
Fuel Capacity, gal	806
Marine Propulsor	(2) propellers; D = 24 in., P/D = 1.3, BAR = 0.945; Retractable
Suspension	Unsprung rigid system, 4 × 4
Brake System	(4) Spot discs
Tires	18 × 25 in., 12-ply
Land Steering	Hydraulic, 2-wheel only
Marine Steering	Hydraulic-actuated twin rudders
Electrical System	24-V DC, single-wire
Hydraulic System	3,000 psi 15 gpm at 2,000 rpm; 3,000 psi 33 gpm at 2,400 rpm
Crew	2
Hull	Welded Aluminum 5456
Planing Surface	Retractable plate with mounted propeller

TABLE 9-3 POWER LIMITED SPEEDS OF LVW PLANING HULL AMPHIBIAN

Sea State	0	1	2	3
Speed, mph	41	40	35.5	26.5

TABLE 9-4 WIND AND SEA SCALE FOR FULLY ARISEN SEA

SEA-GENERAL		WIND				SEA								
						WAVE HEIGHT FEET								
Sea State	Description	(Beaufort) Wind Force	Description	Range (Knots)	Wind Velocity	Average	Significant	Average 1/10 Highest	Significant Range of Periods (Seconds)	T_{max} Period of Maximum Energy of Spectrum	$\bar{T}$ (Average Period)	(Average Wave Length)	Minimum Fetch (Nautical Miles)	Minimum Duration (Hours)
	Sea like a mirror	0	Calm	Less than 1	0	0	0	0	-	-	-	-	-	
0	Ripples with the appearance of scales are formed, but without foam crests.	1	Light Airs	1-3	2	0.05	0.08	0.10	up to 1.2 sec	0.7	0.5	10 in.	5	18 min
1	Small Wavelets, still short but more pronounced; crests have a glassy appearance, but do not break	2	Light Breeze	4-6	5	0.18	0.29	0.37	0.4 - 2.8	2.0	1.4	6.7 ft	8	39 min
	Large wavelets, crests begin to break. Foam of glassy appearance. Perhaps scattered white horses.	3	Gentle Breeze	7-10	8.5	0.6	1.0	1.2	0.8 - 5.0	3.4	2.4	20	9.8	1.7 hr
2					10	0.88	1.4	1.8	1.0 - 6.0	4	2.9	27	10	2.4
	Small waves, becoming larger; fairly frequent white horses	4	Moderate Breeze	11-16	12	1.4	2.2	2.8	1.0 - 7.0	4.8	3.4	40	18	3.8
					13.5	1.8	2.9	3.7	1.4 - 7.6	5.4	3.9	52	24	4.8
					14	2.0	3.3	4.2	1.5 - 7.8	5.6	4.0	59	28	5.2
3					16	2.9	4.6	5.8	2.0 - 8.8	6.5	4.6	71	40	6.6
4	Moderate waves, taking a more pronounced long form; many white horses are formed. (Chance of some spray).	5	Fresh Breeze	17-21	18	3.8	6.1	7.8	2.5 - 10.0	7.2	5.1	90	55	8.3
					19	4.3	6.9	8.7	2.8 - 10.6	7.7	5.4	99	65	9.2
5					20	5.0	8.0	10	3.0 - 11.1	8.1	5.7	111	75	10
	Large waves begin to form; the white foam crests are more extensive everywhere. (Probably some spray).	6	Strong Breeze	22-27	22	6.4	10	13	3.4 - 12.2	8.9	6.3	134	100	12
					24	7.9	12	16	3.7 - 13.5	9.7	6.8	160	130	14
6					24.5	8.2	13	17	3.8 - 13.6	9.9	7.0	164	140	15
					26	9.6	15	20	4.0 - 14.5	10.5	7.4	188	180	17
	Sea heaps up and white foam from breaking waves begins to be blown in streaks along the direction of the wind. (Spindrift begins to be seen).	7	Moderate Gale	28-33	28	11	18	23	4.5 - 15.5	11.3	7.9	212	230	20
					30	14	22	28	4.7 - 16.7	12.1	8.6	250	280	23
					24.5	14	23	29	4.8 - 17.0	12.4	8.7	258	290	24
					32	16	26	33	5.0 - 17.5	12.9	9.1	285	340	27
7	Moderately high waves of greater length; edges of crests break into spindrift. The foam is blown in well marked streaks along the direction of the wind. Spray affects visibility.	8	Fresh Gale	34-40	34	19	30	38	5.5 - 18.5	13.6	9.7	322	420	30
					36	21	35	44	5.8 - 19.7	14.5	10.3	363	500	34
					37	23	37	46.7	6.0 - 20.5	14.9	10.5	376	530	37
					38	25	40	50	6.2 - 20.8	15.4	10.7	392	600	38
					40	28	45	58	6.5 - 21.7	16.1	11.4	444	710	42
8														
	High waves. Dense streaks of foam along the direction of the wind. Sea begins to roll. Visibility affected.	9	Strong Gale	41-47	42	31	50	64	7 - 23	17.0	12.0	492	830	47
					44	36	58	73	7 - 24.2	17.7	12.5	534	960	52
					46	40	64	81	7 - 25	18.6	13.1	590	1100	57

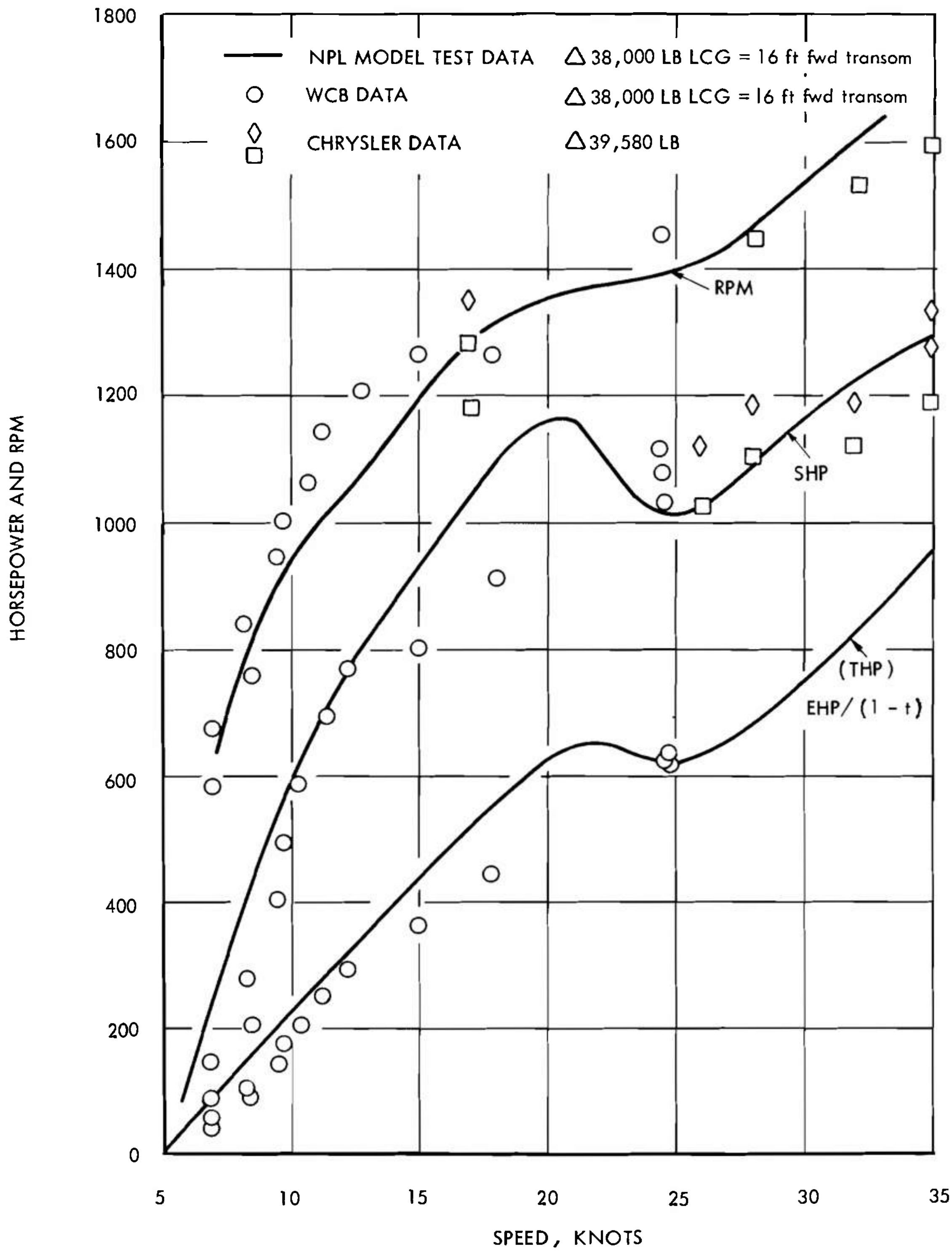


Figure 9-4. LVW-X1 Speed and Power; Model Test and Full Scale Data

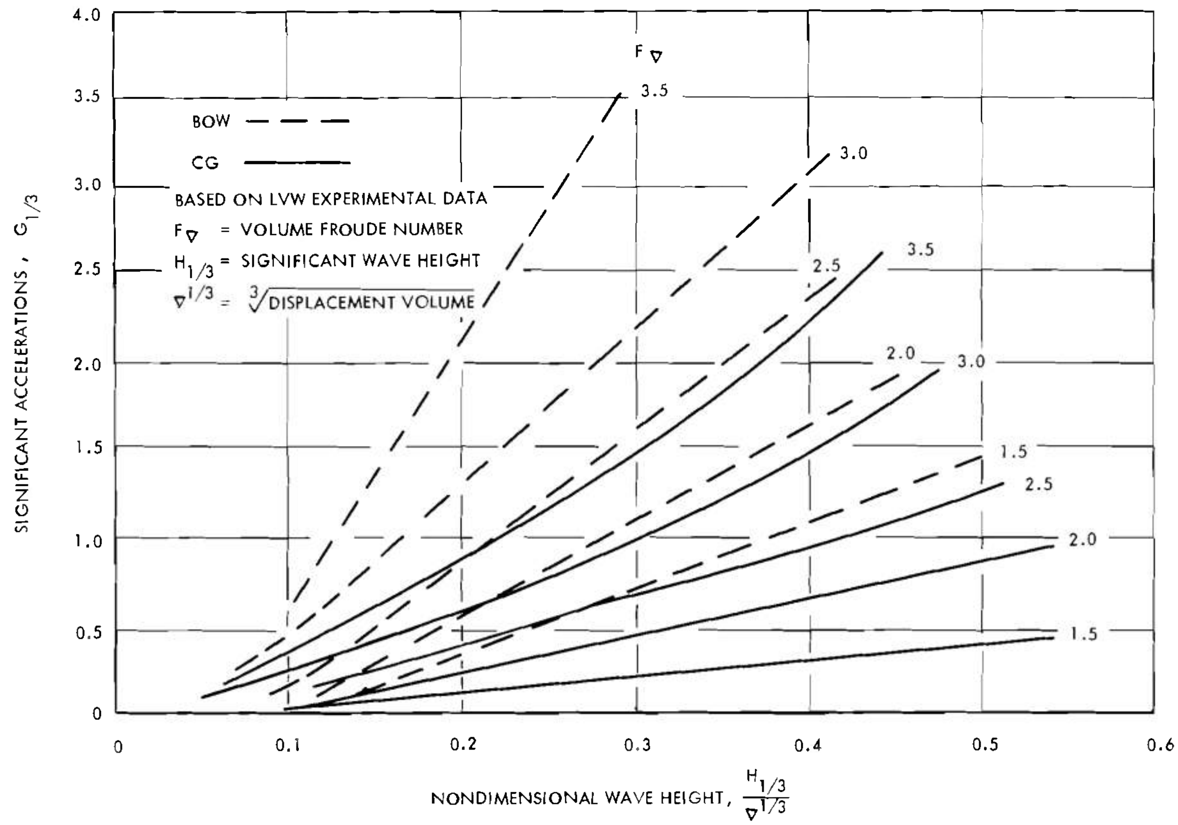


Figure 9-5. "V" Bottom Planing Hull Vertical Accelerations for LVW-X2 in Head Seas

Table 9-7 indicates the considerable increase in sustained speeds in high sea states that will result if the crew is located in the area of minimum accelerations aft of the center of gravity.

9-2.1.3 Control

The control characteristics of the LVW-X2 were measured at high and low speeds, and the results reported in Ref. 5. The observed turning diameters at high and low speeds are presented in Tables 9-8 and 9-9, respectively.

In high-speed turns the LVW heels inboard and in general shows satisfactory stability characteristics. Fig. 9-7 presents the arrangement of the propellers and rudders on the LVW.

Because of the very low reliability of the LVW steering and suspension systems, only limited data are available on the control characteristics on land. Ref. 6 indicates turning diameters of 138 ft to the right and 120 ft to the left with 2-wheel steering. No tests were conducted with 4-wheel steering.

9-2.2 FUTURE PLANING HULL AMPHIBIANS

From data obtained with the LVW-X1 and LVW-X2 and other studies of planing hull performance, it is possible to make some projections of the possible characteristics and performance of future planing hull amphibians. Projections of this type have been made and are reported in Refs. 4 and 7.

9-2.2.1 Drag

Experimental studies (Refs. 8 and 9) indicate that more recent "V" bottom and inverted "V" bottom planing hulls with sizes and displacements suitable for amphibians can be designed with less resistance than the LVW-X1 and LVW-X2. A comparison between the drag of the LVW-X1 and that of a Series 62 "V" bottom hull and LVW-X3 inverted "V" bottom hull is

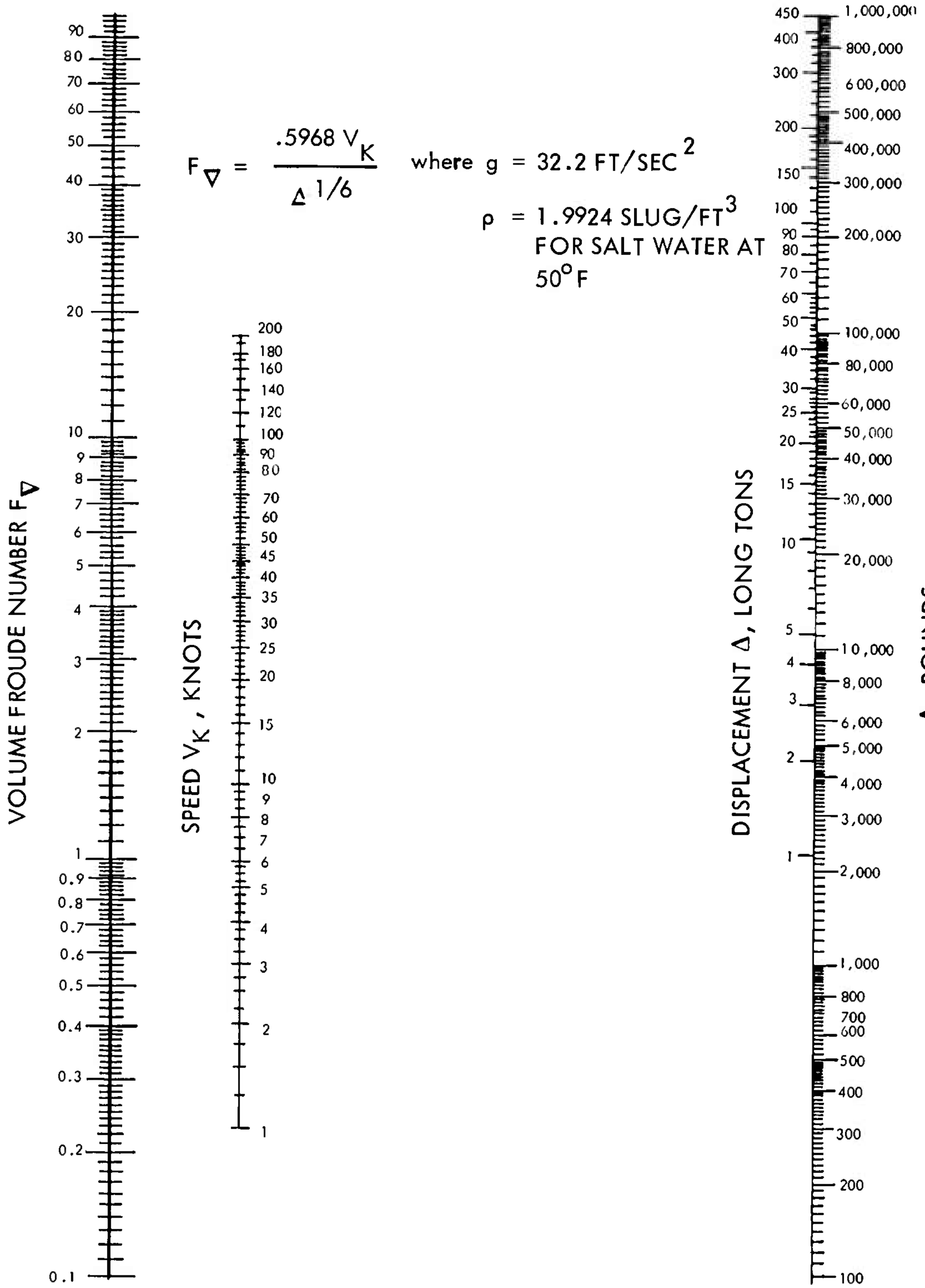


Figure 9-6. Nomograph of Speed, Displacement, and Froude Number

TABLE 9-5 PLANING HULL ACCELERATIONS IN FOLLOWING SEAS

Volume Froude Number F_∇	*Acceleration (as Percentage of Acceleration in Head Seas)*
1.5	20%
2.0	21%
2.5	29%
3.0	33%
3.5	43%
4.0	64%
4.5	79%

TABLE 9-6 SUBJECTIVE REACTIONS OF PERSONNEL TO HIGH-SPEED AMPHIBIAN MOTIONS

Significant Upward Acceleration $G_{1/3}$	*Reaction*
0.35	Motion Inconsequential
0.9	Tolerance Level = 1 hr
1.3	Tolerance Level = 0.5 to 1 hr
2.1	Tolerance Level = 10 to 15 min
2.3-2.6	Beyond Limit of Endurance

TABLE 9-7 SUSTAINED SPEED OF LVW IN HEAD SEAS GROSS WEIGHT 39,500 POUNDS

	Crew Forward				*Crew Aft*			
Sea State	0	1	2	3	0	1	2	3
Speed, mph	41	39	25	12	41	40	35	24

presented in Fig. 9-8. The sizes and displacements indicated would be suitable for an amphibian with a 10,000-lb payload capability. Figs. 9-9 and 9-10 present the characteristics of the Series 62 and LVW-X3 hull forms, respectively. Neither of these hull forms has wheel wells, and thus the forms are representative of amphibians with completely faired wheel wells. In practice it will be very difficult to provide covers for the forward wheels because of the high hydrodynamic loads in rough water. Based on data in Ref. 3 an estimate was made of the drag of a "V" bottom planing hull form with open bow wheel wells and completely covered aft wheel wells. These data are also included in Fig. 9-3. As expected, the drag of such a form is between that of the completely faired hulls and the LVW-X1 with open bow and stern wheel wells.

For the purpose of a very preliminary estimate of the drag of a planing hull amphibian design, the data in Fig. 9-8 may be replotted in the form of a drag-weight ratio as a function of volume Froude number. A new planing hull amphibian may be assumed to have the same drag-weight ratio at the same volume Froude number. As soon as possible, however, a more exact drag estimate that accounts for differences in hull form, displacement, and center of gravity location should be prepared. Refs. 3, 8, 9, and 10 may be used for this purpose.

For planing hulls, the drag in waves is often estimated by a percentage increase over calm water drag. The percentage used must be based on experience. Fig. 9-11 presents the percentage drag increase in rough water as a function of nondimensional wave height and volume Froude number. Data are included from tests of several planing hull forms whose characteristics are given in Ref. 9, 11, and 12. The data in Fig. 9-11 may be used with the calm water drag data previously discussed to obtain the estimate of the drag of a future planing hull amphibian in waves.

The power required for a high-speed planing hull amphibian may be determined once the drag is estimated by using the procedures described in par. 7-7.

9-2.2.2 Accelerations in Waves

In considering future planing hull amphibians, it will be necessary to predict their acceleration characteristics. The data presented in Fig. 9-5 for the LVW-X2 may be used for this purpose. A comparison between the acceleration of the LVW-X2 and other planing hulls has been made and the results are presented in Fig. 9-12. The accelerations of the LVW-X2 follow a similar trend with speed as that of the other planing hulls considered. The deadrise angle (at a location 30 percent of the length aft of the bow) has a significant effect on the accelerations. This is consistent with the expected effect of deadrise as reported in Ref. 13. It is of interest to note that the accelerations of the LVW-X3 inverted "V" hull are similar to those of the "V" bottom hulls with deadrise angles between 25 and 30

TABLE 9-8 LVW-X2 HIGH-SPEED TURNING DIAMETERS

Direction	*Diameter, ft*	*Time, sec*	*Average Speed, mph*	*Turn Diameter/Vehicle Length*
Starboard	720	53	29.0	20
Port	640	53	26	17.8
Starboard	720	53	29	20
Port	670	53	27	18.6
Starboard	710	52	29	19.7
Port	690	54	27	19.2
Starboard	780	55	30	21.7
Port	870	58	32	24.2

TABLE 9-9 LVW-X2 SURF MODE TURNING DIAMETERS

Direction	*Speed, mph*	*Diameter, ft*	*Trim,*(1) *deg*	*Heel, deg*	*Turn Diameter/Vehicle Length*
Port	5.5(2)	81	4	2.5 Stbd	2.25
Starboard	5.5	95	4	7 Port	2.64
Port	6.6	91	4.5	4 Stbd	2.52
Starboard	6.6	100	4.5	8 Port	2.78
Port	8.5	104	6	7 Stbd	2.89
Starboard	8.5	136	6	11 Port	3.78
Port	10.5(3)	150	9	9 Stbd	4.16
Starboard	10.5	140	9	14 Port	3.88

NOTES:

(1) Trim by the stern; Static trim = 3 deg; *GVW* = 39,500 lb; Four (4) personnel on board; *KG* = 3.2 ft; Load *VCG* approximately 20 in. above deck

(2) Minimum speed dictated by high turbine exhaust temperature

(3) Maximum speed in surf mode

deg. Based on the above comparisons, it is concluded that the accelerations of the LVW-X2 are typical of what can be expected of a future planing hull amphibian.

9-2.2.3 Vehicle Characteristics

Ref. 7 presents a parametric study of high-speed amphibian characteristics. The configuration of the base planing hull amphibian developed in this study is shown in Fig. 9-13. Parametric variations were made of this configuration to determine the effects of payload and range on vehicle characteristics. The results in terms of gross vehicle weight, power, principal dimensions, and payload fraction—all as a function of payload—are presented in Fig. 9-14. All of these characteristics are for craft

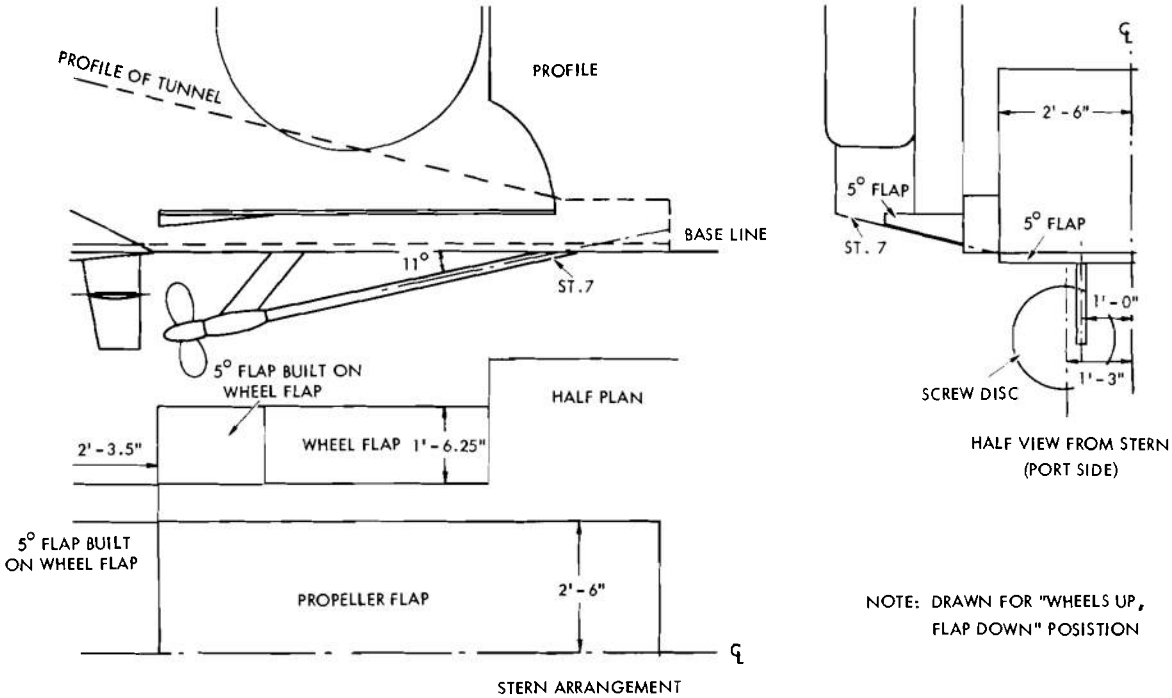


Figure 9-7. LVW-X1 Stern Arrangement

with power limited speeds of 30 knots in Sea State 2. This performance was estimated on the basis of data similar to those presented in Fig. 9-8 for a vehicle with open bow wheel wells and faired aft wheel wells. The increased drag due to waves was determined from the data presented in Fig. 9-11. The craft were assumed to be propelled with a gas turbine engine with a specific weight of 2 lb/hp and a fuel rate of 0.75 lb/hp-hr. The vehicle characteristics presented in Fig. 9-14 are believed to reflect the current (1968) state-of-the-art in planing hull amphibians.

9-3 HYDROFOIL AMPHIBIANS

Another vehicle type which may be suitable for use as a high-speed amphibian is the hydrofoil. On a hydrofoil amphibian at high speed, foils in the water supply sufficient lift to support the main hull above the water surface. This eliminates the large wave drag associated with displacement-type amphibians, making high speeds possible at reasonable power levels. As in the case of a planing hull amphibian, a hydrofoil amphibian may require higher power at an intermediate speed. This may occur at speeds between 12 and 25 mph before the hull lifts

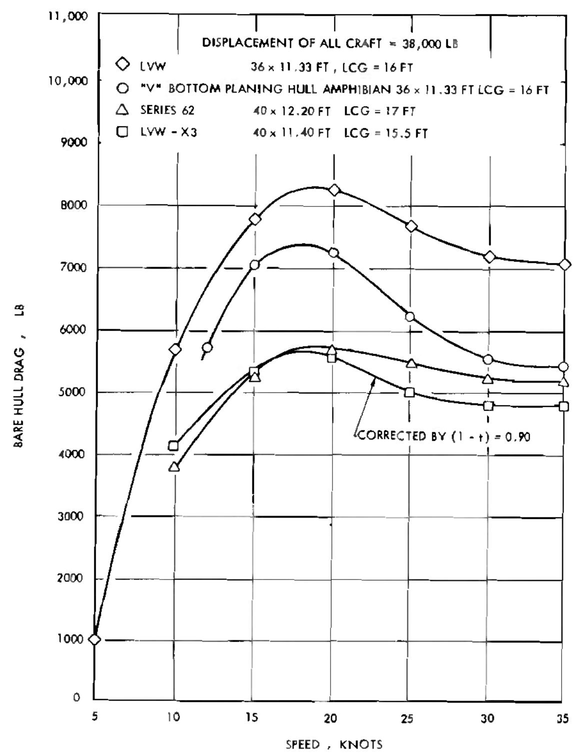
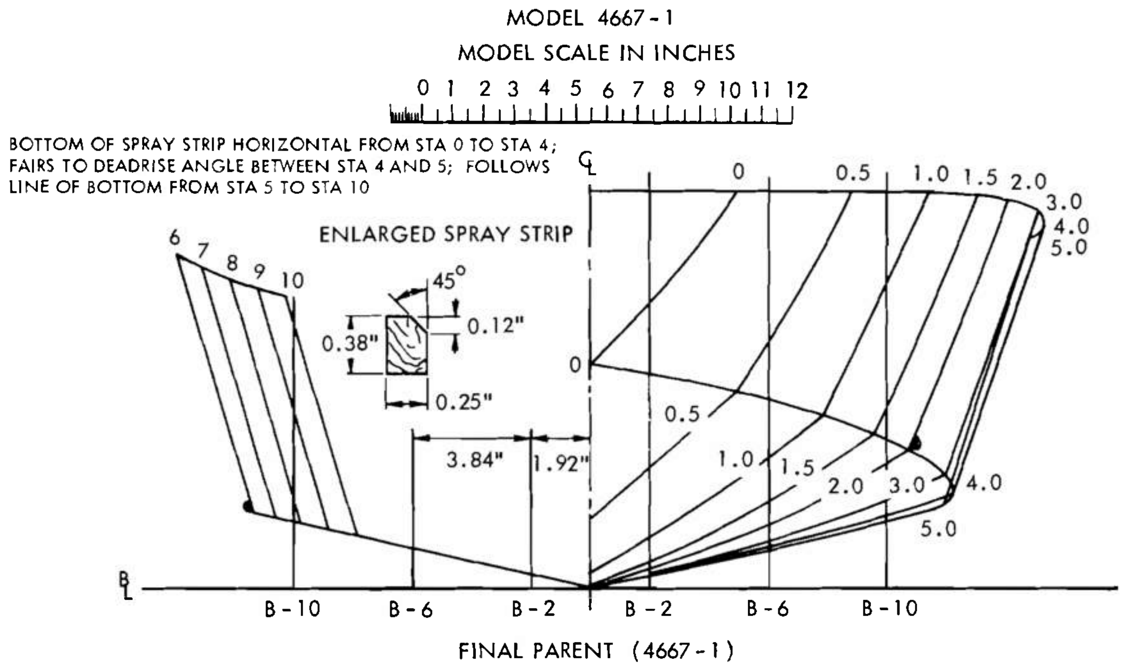
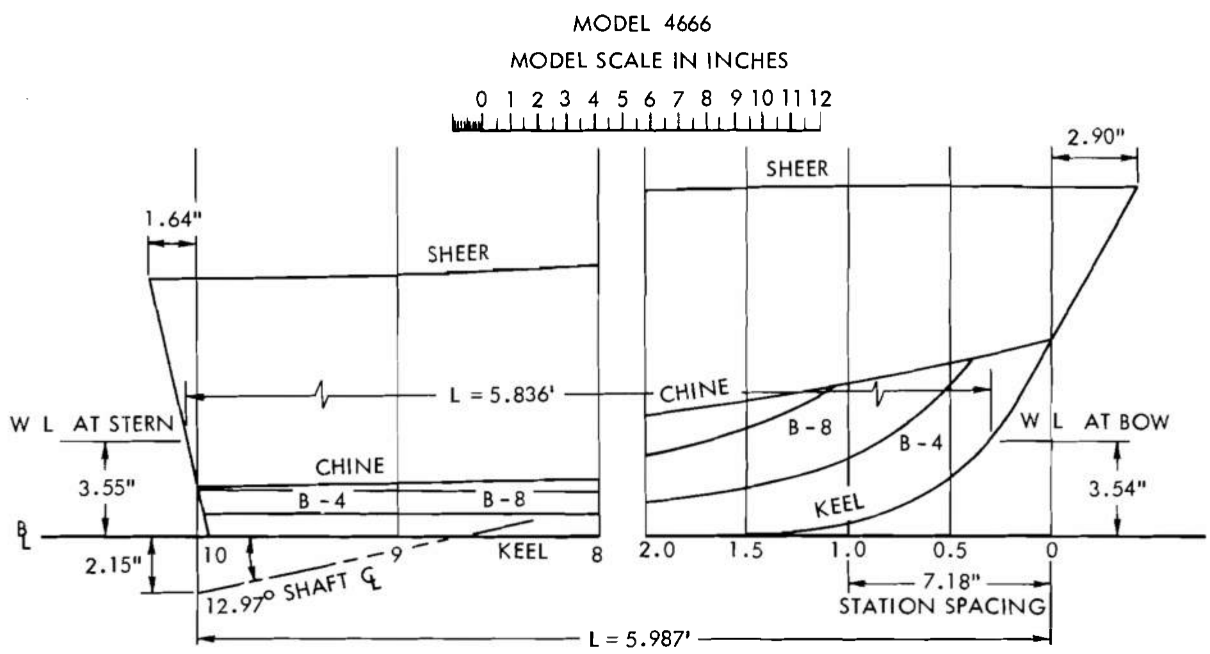


Figure 9-8. Comparison of Planing Hull Amphibian Drag



BODY PLAN OF SERIES 62 PARENT MODEL



BOW AND STERN ENDINGS OF SERIES 62 MODEL

Figure 9-9. Series 62 Lines

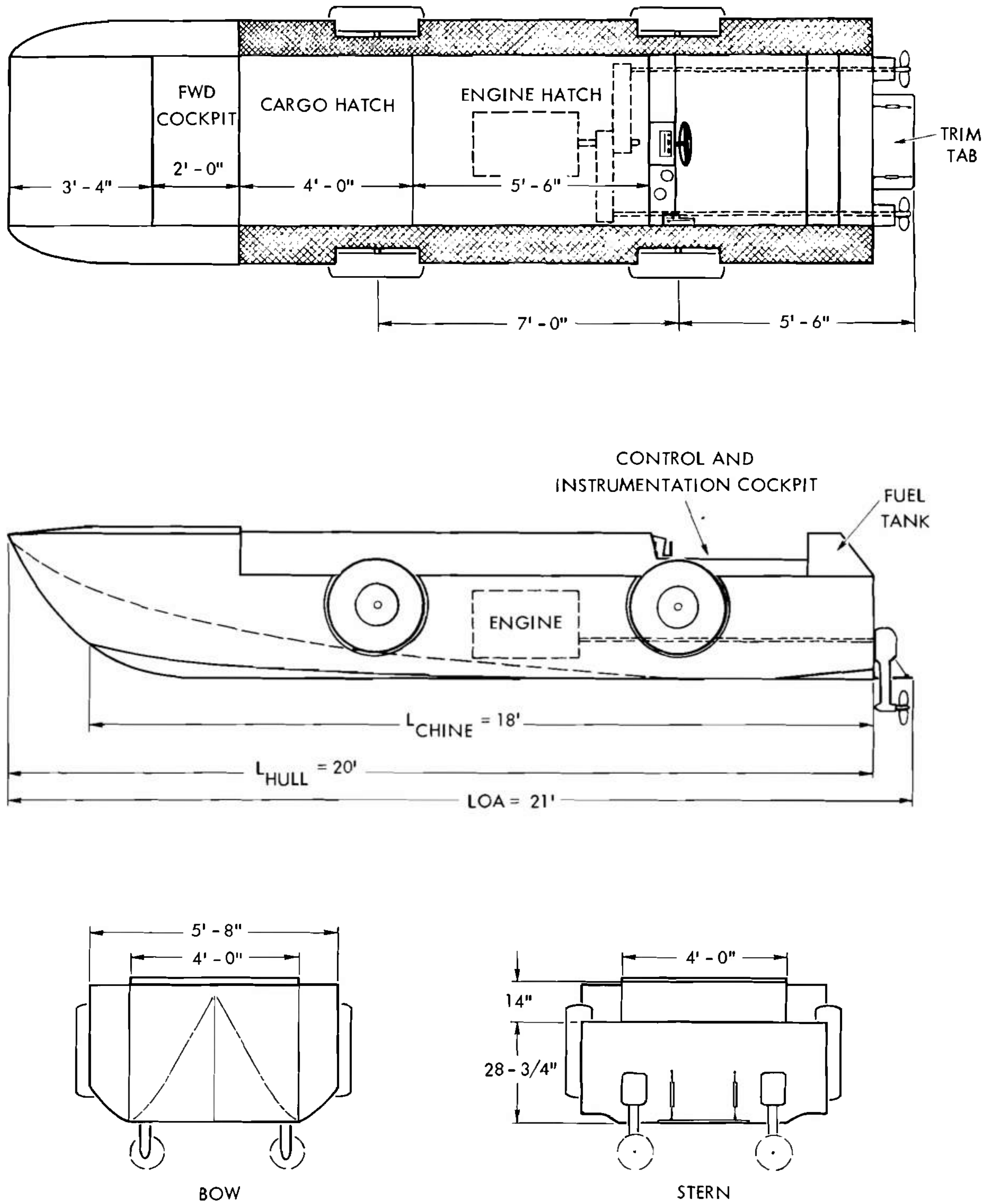


Figure 9-10. General Arrangement of LVW-X3 (Model)

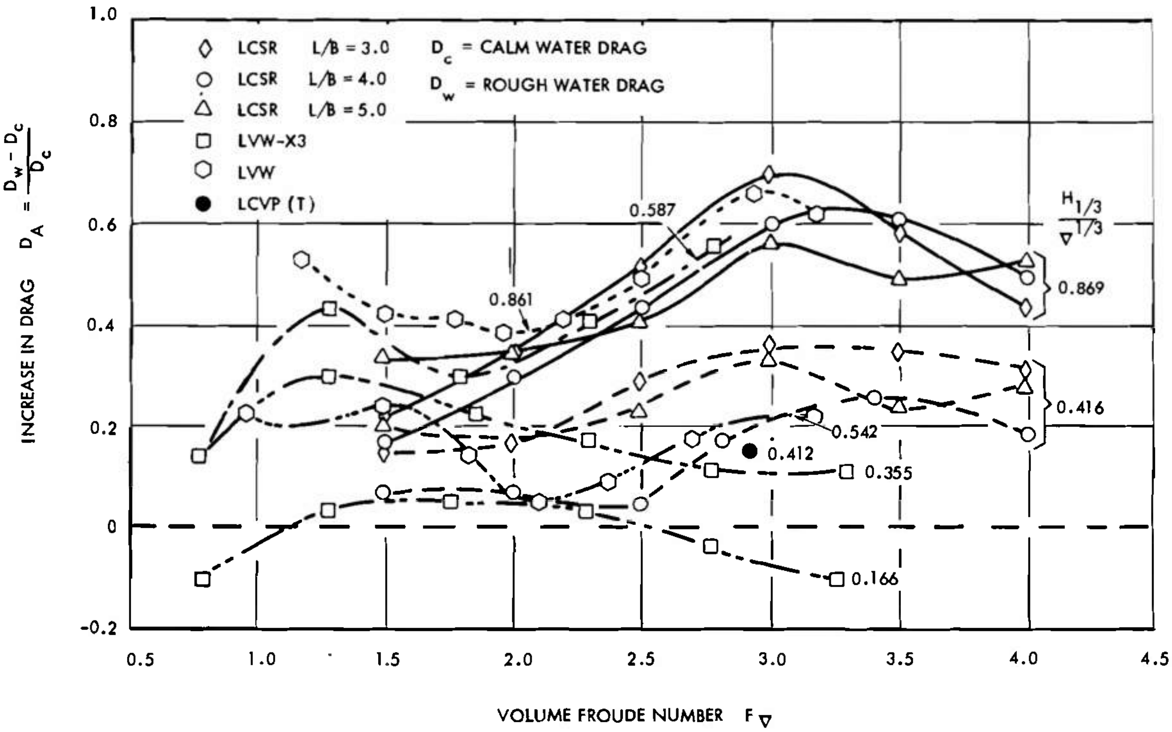


Figure 9-11. Planing Hull Added Drag in Waves

clear of the water. This problem is discussed in par. 9-3.1.4.1. Other problems with hydrofoil amphibians include the difficulty of foil retraction and structural reinforcement in way of the foils.

9-3.1 EXISTING HYDROFOIL AMPHIBIANS

The concept of a hydrofoil high speed amphibian was first proposed in the middle 1950's. Work on craft of this type was first sponsored by the U. S. Army Ordnance Corps and later by the Navy Bureau of Ships for the U. S. Marine Corps. As a result of these efforts three vehicle types were built. These include the "Flying DUKW", the LVH-X1, and the LVH-X2.

9-3.1.1 The Flying DUKW

In April of 1956 the Lycoming Division of the AVCO Corporation undertook a feasibility study for the U. S. Army Ordnance Corps to determine from theoretical and towing tank studies whether a WW II DUKW could be flown on hydrofoils. This study showed that it would be possible to do this and a contract was let to so equip a WW II DUKW. The conversion amounted to adding fully submerged hydrofoils, stern drive assembly, Lycoming T-53 gas turbine, and associated equipment. The actual conversion work was done by the Miami Shipbuilding Corporation. The basic conversion was completed in 1959 and the craft was reported to have operated at a gross weight of 29,600 lb and at a water speed in excess of 34 mph. Fig. 9-15 is a photograph of the Flying DUKW foilborne. The Flying DUKW experienced difficulty in becoming foilborne in waves because of increased drag.

The Flying DUKW was not intended to be a prototype for an operational vehicle. The addition of the foils, stern drive, and gas turbine engine more than used up the weight and volume available for cargo. Its purpose was to act as a test bed craft which would prove the feasibility of a hydrofoil amphibian and provide data for its design.

9-3.1.2 The LVH-X1

The U. S. Marine Corps Landing Force

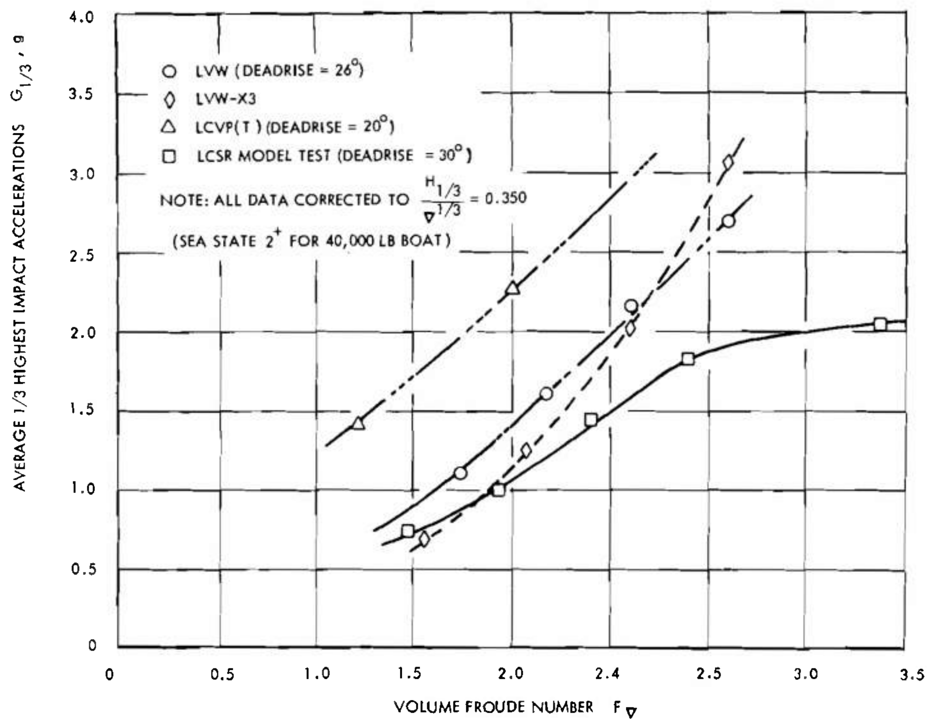


Figure 9-12. Planing Hull Accelerations at Bow (1 of 2)

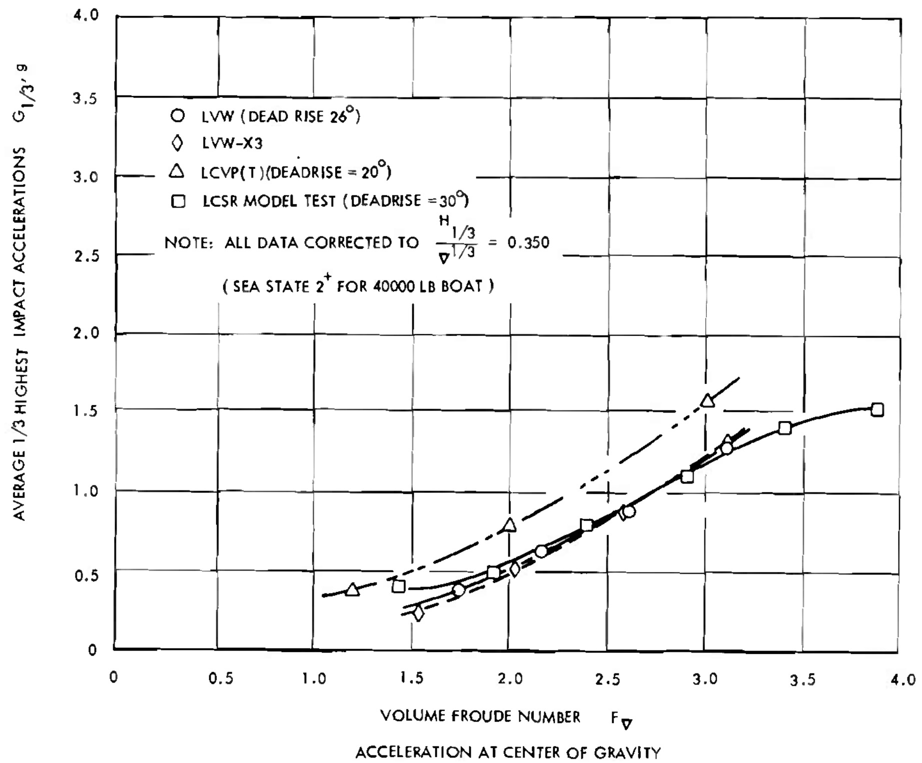


Figure 9-12. Planing Hull Accelerations at Bow (2 of 2)

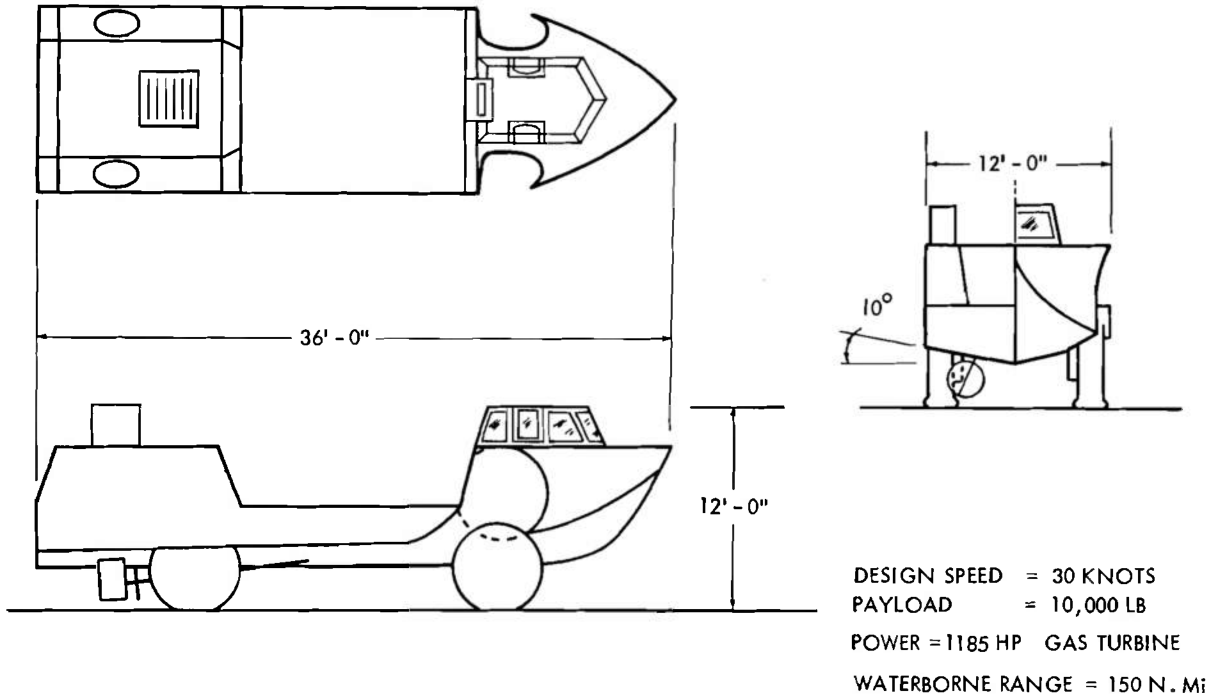


Figure 9-13. High-speed Amphibian—Planing Hull Type, Base Vehicle Configuration

Development Center was assigned responsibility for coordination of the testing and evaluation of a hydrofoil amphibian (LVH). Contracts were awarded by the Navy Bureau of Ships for the design and construction of two different hydrofoil amphibians. The first contract was awarded to the Lycoming Division of AVCO Corp. and was designated LVH-X1. Two identical vehicles were constructed. The vehicle concept that became the LVH-X1 was first proposed by Lycoming in 1959. Actual design of the LVH-X1 started in 1961 and the two vehicles were accepted in late 1964 and early 1965, respectively.

The foil system used in the LVH-X1 was based on fully submerged incidence-controlled foils. This type of foil system has no inherent stability and depends on an auto-pilot to control flying height and attitude. The advantage of this type of foil system is that very low motions in waves are possible. In the foil mode, additional control forces are generated by flaps on the forward foils controlled by the auto-pilot. A sonic height sensor was used to provide the auto-pilot with information on foil immersion.

9-3.1.2.1 Characteristics

The LVH-X1 was powered with a TF-1460-B1A gas turbine. This engine was a marine conversion of an aircraft gas turbine with a continuous rating of 1,000 *BHP* and an intermittent rating of 1,225 *BHP* on an 80°F day. The hydrofoils were supported on two hydraulically actuated rotating struts. Thrust for operation in the foil mode was supplied by a conventional propeller mounted on the forward end of a pod on the aft strut assembly. For operation in the displacement mode, the LVH-X1 was propelled by a single retractable directionally controllable propeller in a Kort nozzle.

For land operation, the vehicle was equipped for 2- and 4-wheel drive and hydraulically controlled steering of the forward wheels. The power train had an overall ratio (engine to wheel) in low gear of 194.6:1 and in high gear of 68.9:1. The transmission was a 2-speed mechanical dry-sump type. Service brakes were disc-type, hydraulic-actuated. Parking brakes were the internal-expanding-type, mounted to

the forward differential shaft. All wheels retracted hydraulically into the hull for the boating and flying modes, and to facilitate ease of loading/unloading in the land mode.

The LVH-X1 was designed to carry a payload of 10,000 lb on land and water. The design water speed was 30 mph and the design land speed was 40 mph.

Figs. 9-16 and 9-17 present photographs of the LVH-X1 operating on land and water, respectively. The characteristics of the LVH-X1 are presented in Table 9-10 and the external arrangement in Fig. 9-18.

9-3.1.2.2 Performance of LVH-X1

No engineering data are available on the high-speed waterborne performance of the LVH-X1. Because of reliability problems, the two LVH-X1 vehicles never underwent engineering tests to determine their foilborne performance. All testing of the LVH-X1 vehicle stopped when one sank after the failure of a drive shaft in foilborne operation.

Some engineering tests were conducted on the LVH-X1 in the displacement mode and are reported in Refs. 14 and 15. The most important finding of these tests was that the trainable propeller used for hullborne steering provided excellent control. The tactical diameters were reported to be 121 ft to starboard and 63 ft to port in the surf mode. Because of the high rate at which water leaked into the vehicle, it was not possible to conduct an inclining experiment and determine the static stability.

Land performance tests for the LVH-X1 are reported in Ref. 16. It was concluded that the performance of the vehicle under desert conditions was satisfactory but deficiencies were encountered with respect to vehicle durability and/or reliability. The maximum speed of the vehicle was 45 mph and it was successfully driven up a 60 percent slope in both forward and reverse gear. The vehicle was not safe on a 30 percent side slope. The minimum turning radii were 48 ft to the right and 53 ft to the left.

9-3.1.3 The LVH-X2

A second contract for the design and construction of a hydrofoil amphibian was awarded by the Navy Bureau of Ships to the FMC (Food Machinery Corp.) Corporation. Two identical vehicles were built and were designated LVH-X2. The design of the LVH-X2 began in 1961, and the two vehicles were accepted in late 1963 and early 1964.

The foil system used on the LVH-X2 is based on a surface-piercing foil forward and a fully submerged foil aft. This foil system has inherent stability in heave, pitch, and roll so that foilborne operation without an auto-pilot is possible in calm water. For foilborne operation in high sea states (Sea State 3), an auto-pilot is required. The auto-pilot on the LVH-X2 uses a sonic height sensor and exerts control forces with flaps on the fore and aft foils.

The foil system retracts by allowing the struts to pass vertically into the hull and the foils to pivot or seat in faired recesses in the hull. Retraction is accomplished by hydraulic motor driven ball screw actuators.

9-3.1.3.1 Characteristics

The LVH-X2 is powered with a Solar Saturn gas turbine engine with a continuous rating of 1,050 *BHP*. Power produced by the turbine is delivered through a primary reduction gear to the transfer case which distributes power to the land drive train, the marine drive train, or both, depending on the mode of operation. Land power is transmitted to the wheels through a 6-speed TX 365-2 transmission and differentials.

In the land mode, hydraulically powered 2-wheel, 4-wheel, or oblique steering can be accomplished. Each wheel is retractable using an air hydraulic cylinder which also provides spring and damping action. In all waterborne modes, thrust is provided by a single propeller mounted on the forward end of a pod on the aft foil strut. In the foilborne mode, steering is provided by a rudder mounted on the aft strut. In the displacement condition, this rudder along with the wheels provide the steering forces.

The LVH-X2 was designed to carry a payload of 10,000 lb on both land and water. The foilborne design speed was 35 knots, the boating mode design speed was 12 knots, and the land design speed was 40 mph.

Figs. 9-19 and 9-20 present photographs of the LVH-X2 operating on land and water, respectively. The characteristics of the LVH-X2 are presented in Table 9-11, and the external dimensions and foil retraction arrangement in Figs. 9-21 and 9-22, respectively.

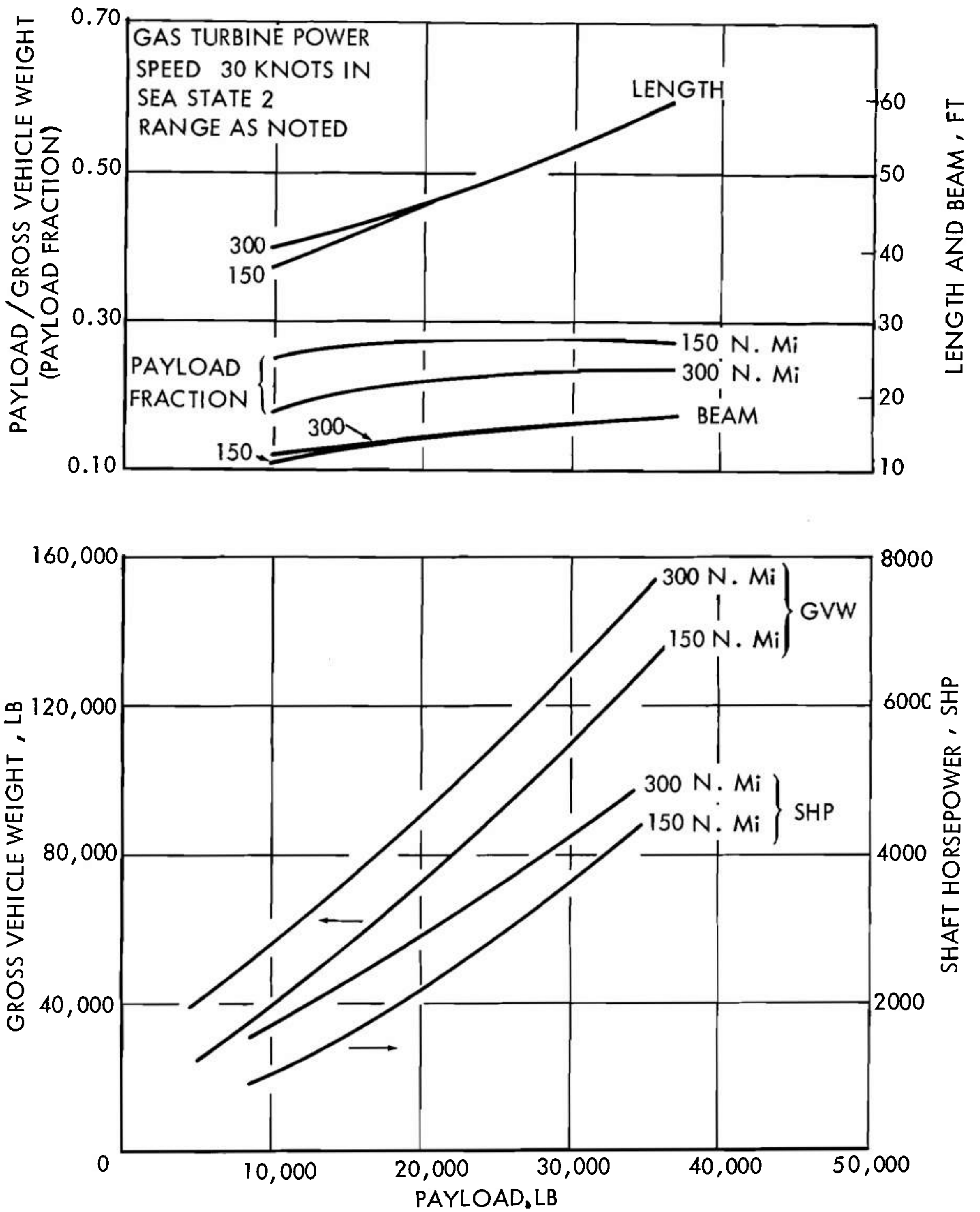


Figure 9-14. High-speed Amphibian—Planing Hull Type, Effect of Payload

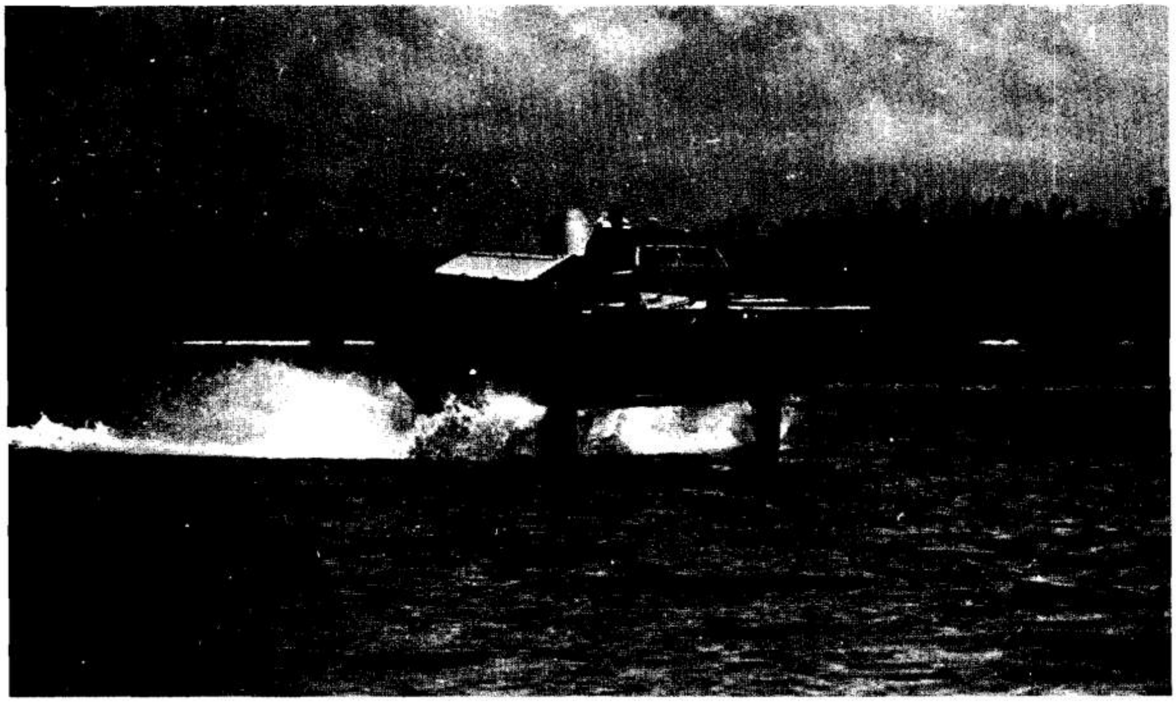


Figure 9-15. Flying DUKW, Foilborne

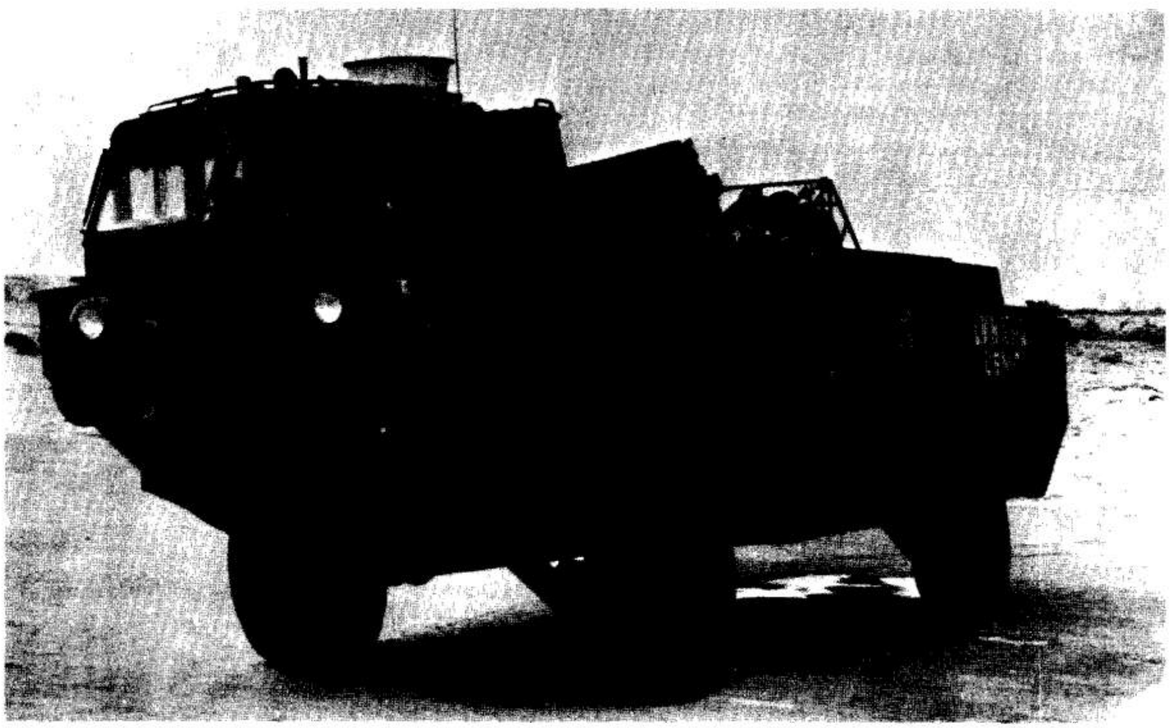


Figure 9-16. LVH-X1 on Land

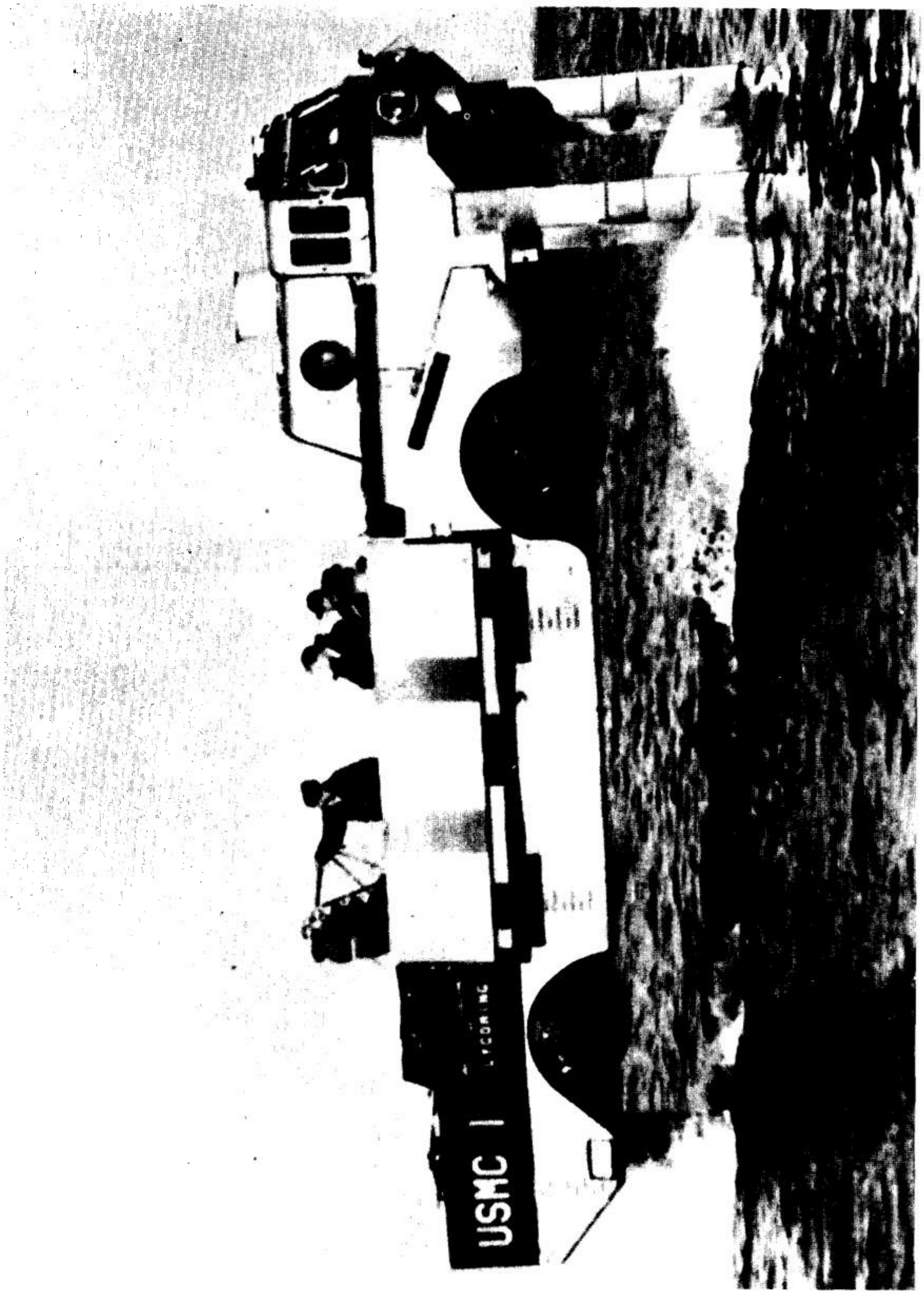


Figure 9-17. LVH-X1 Operating Foilborne

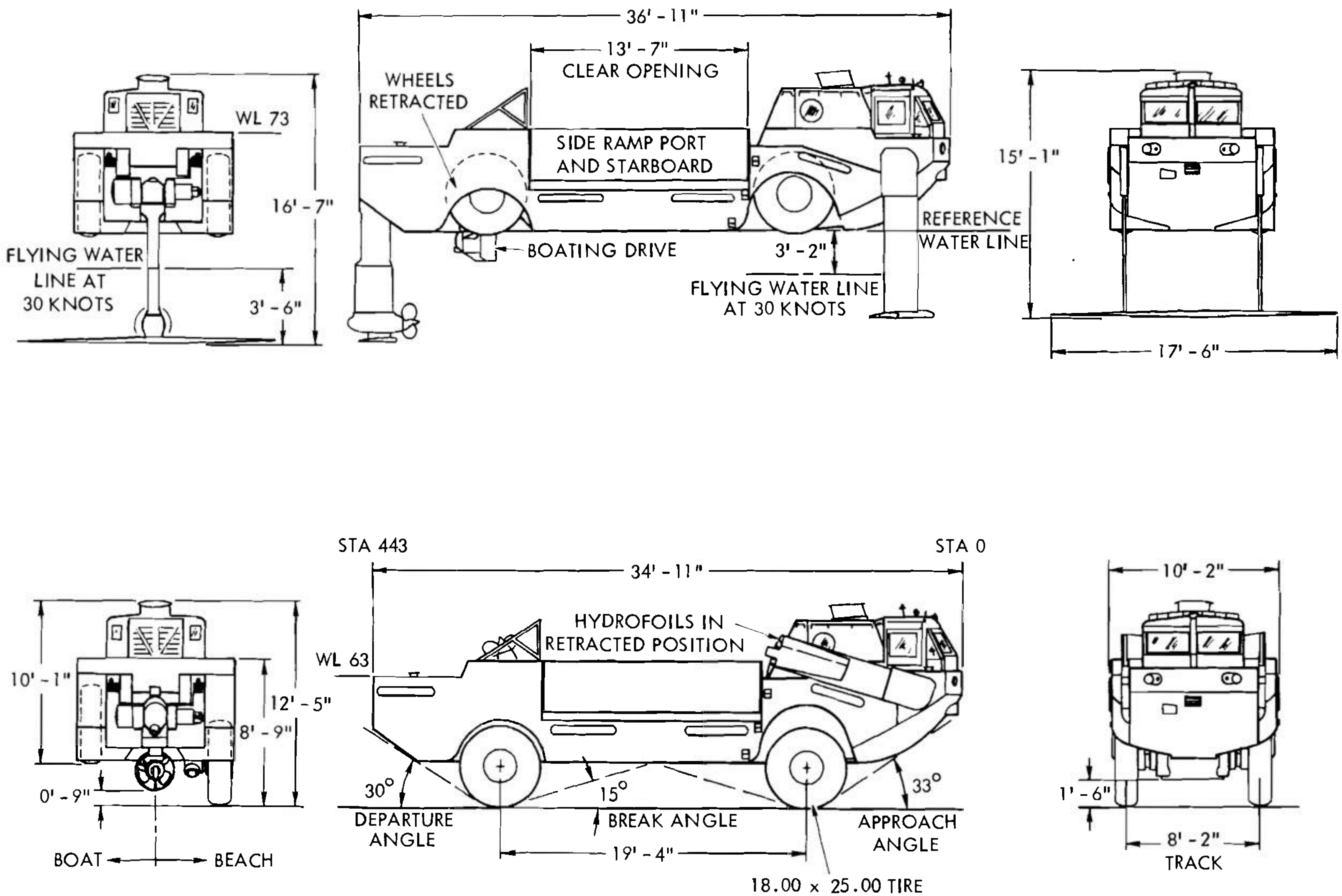


Figure 9-18. External Dimensions of LVH-X1

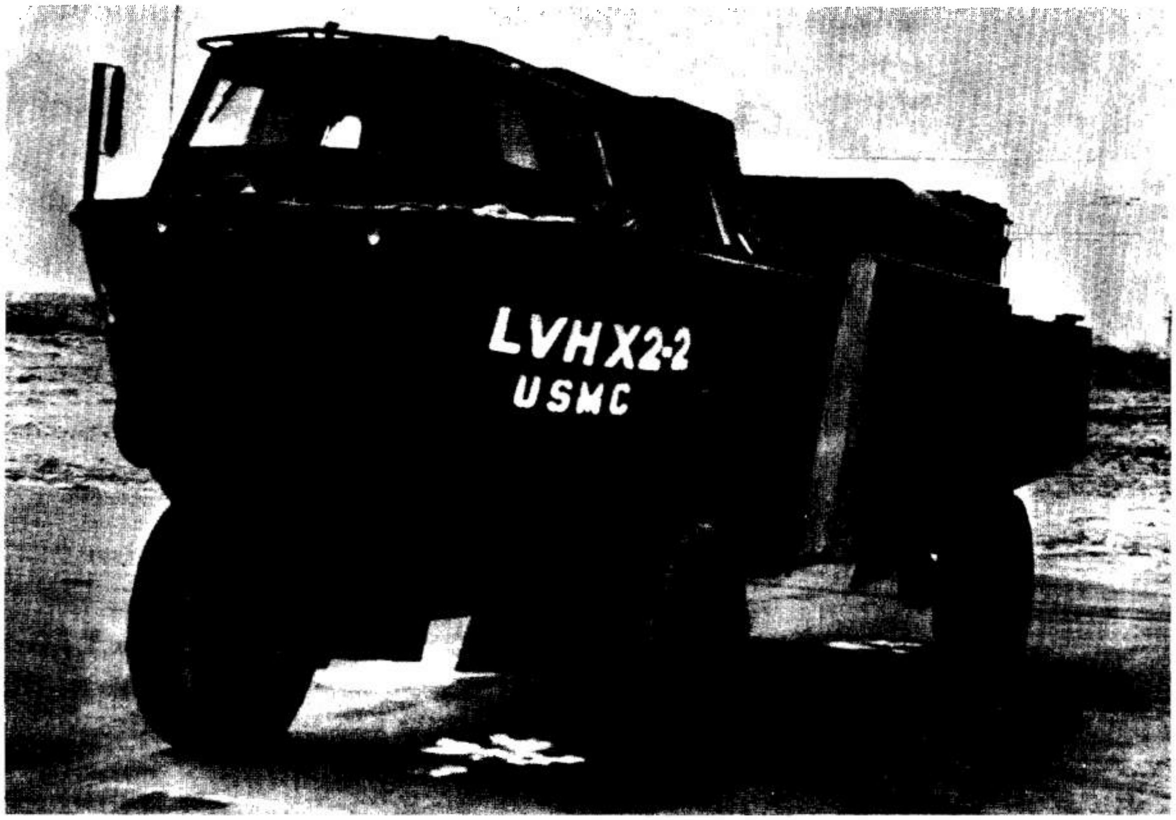


Figure 9-19. LVH-X2 on Land

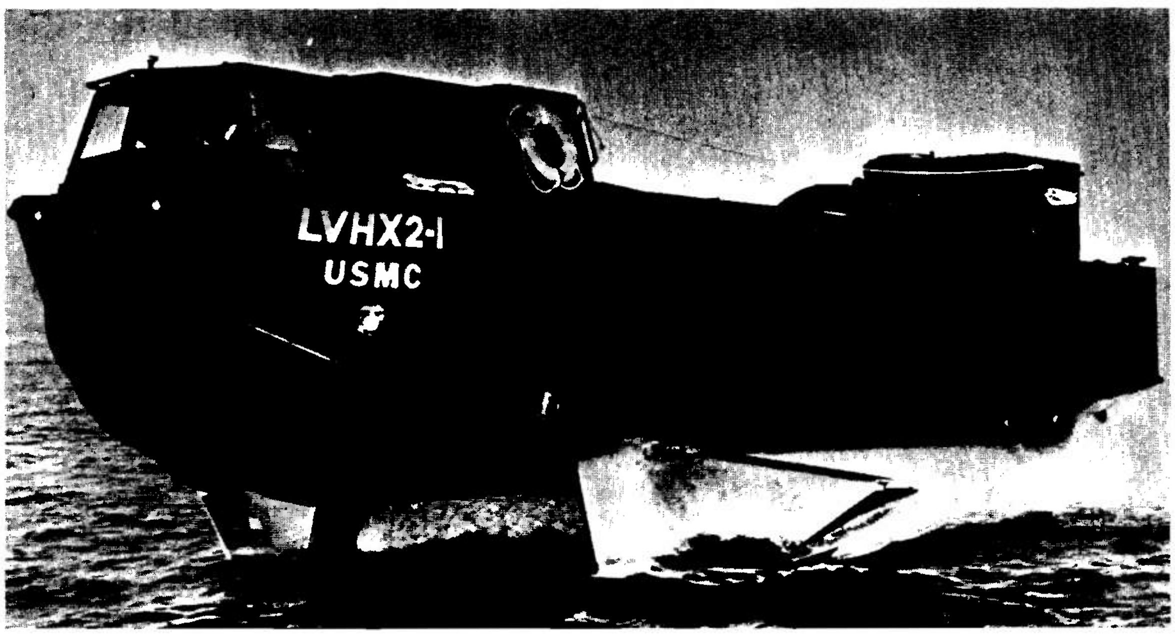


Figure 9-20. LVH-X2 Operating Foilborne

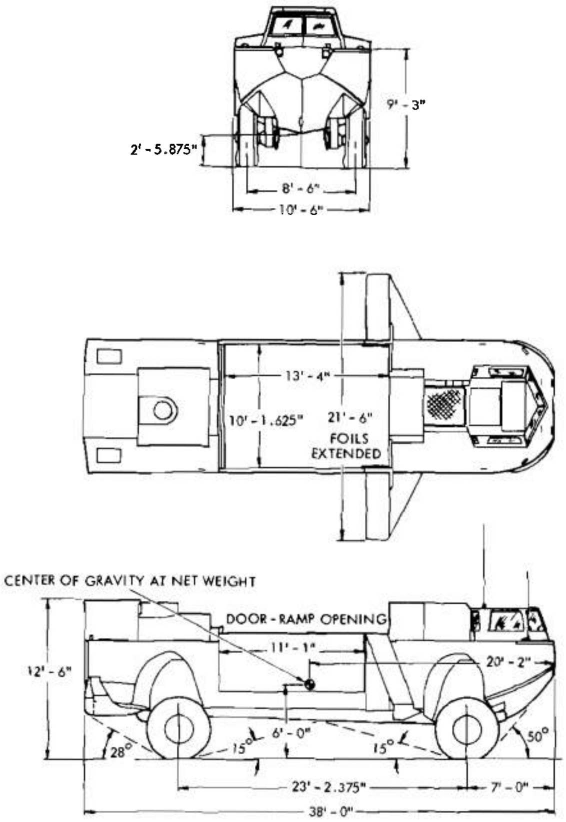


Figure 9-21. External Dimensions of LVH-X2

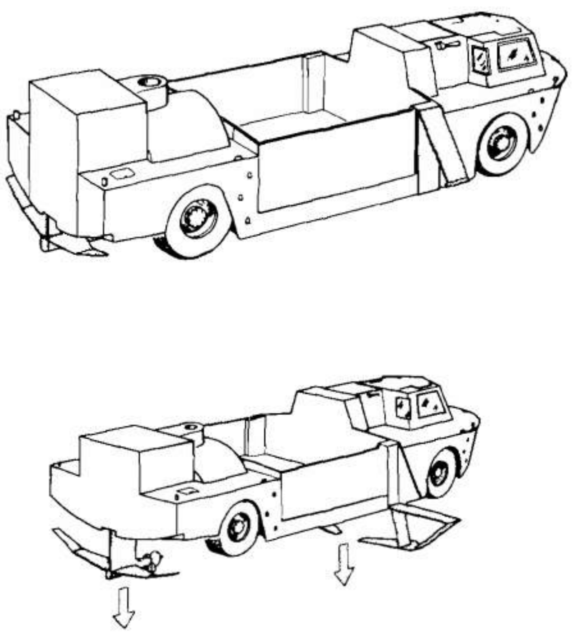


Figure 9-22. Hydrofoil System for LVH-X2

9-3.1.3.2 Waterborne Speed and Power of LVH-X2

As with the case of the LVW planing hull amphibian, the U. S. Marine Corps sponsored a series of engineering tests to determine the waterborne performance characteristics of the LVH-X2. The results of the speed resistance and powering tests are reported in Ref. 17. These tests were conducted over a range of gross weights in calm water and waves. Fig. 9-23 presents a summary of the calm water drag data obtained from Ref. 17. The data labeled deduced drag were obtained from measured thrust during acceleration runs by subtracting the force due to the measured acceleration. These are only approximate since they do not account for any unsteady effects. However, the deduced drag does show a large drag hump at speeds between 10 and 20 knots. This will cause a high power requirement and means that the power available in a hydrofoil amphibian must be checked at take-off as well as the design point.

Fig. 9-24 presents some data on the percentage drag increase in waves. The volume Froude number and nondimensional wave height are the same as defined in par. 9-2.1.2. Also presented in Fig. 9-24 are some data on the added drag in waves of the H. S. DENISON, a large hydrofoil boat with a foil system of similar geometry to that of the LVH-X2.

Based on these data, the top speeds (power limited) of the LVH-X2 in calm water and waves was estimated in Ref. 4. These speeds are given in Table 9-12.

9-3.1.3.3 Accelerations of LVH-X2 in Waves

The LVH-X2 was instrumented and the accelerations in head seas were measured at the bow (crew cab) and at the center of gravity. These data are presented in Ref. 4. Fig. 9-25 presents the average of 1/3 highest upward acceleration in g's for the LVW-X2 as a function of volume Froude number.

If the same 0.9 g allowable acceleration level is assumed as for a planing hull amphibian (see par. 9-2.1.2), a motion limited speed for a hydrofoil amphibian can be defined. This level and the resulting speed and wave heights are included in Fig. 9-25. Some data have also been included for the H.S. DENISON.

The motions of the LVH-X2 are quite different from those of the LVW. The

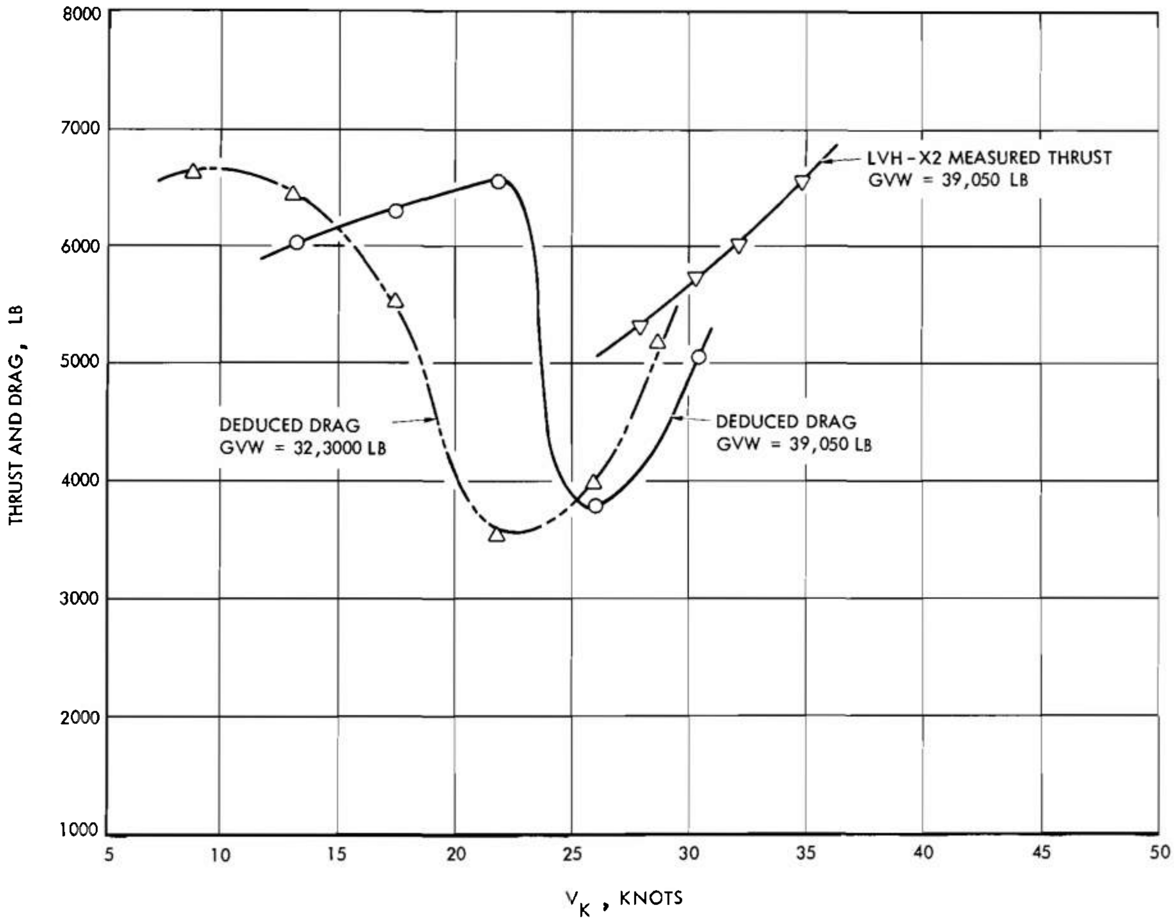


Figure 9-23. Hydrofoil Drag

accelerations are lower and in general there is little difference between the values at the bow and center of gravity. It should be realized that the acceleration of a hydrofoil amphibian will be strongly influenced by the characteristics of the auto-pilot used for control in waves. Also, a hydrofoil amphibian will have difficulty flying in following seas although the acceleration will be low. In actual practice the performance of the LVH-X2 in high waves (Sea State 3) was limited by the ingestion of salt water into the gas turbine. The resulting loss of power made it impossible to operate for long periods. In the design of a new hydrofoil amphibian it should be possible to eliminate this problem (see par. 7-5.2.2).

9-3.1.3.4 Control Characteristics of LVH-X2

The control characteristics of the LVH-X2 were measured at high and low speeds, and the results reported in Ref. 18. The observed turning diameter at high and low speeds are presented in Tables 9-13 and 9-14, respectively.

The natural tendency of a hydrofoil amphibian is to heel outboard in a turn. In the LVH-X2 this tendency is counteracted by the roll control ability of the auto-pilot system. If too tight a flying turn is attempted, the flaps reach full deflection and the increase in drag is sufficient to slow the craft below flying speed. It may be expected that the high-speed control characteristics of a hydrofoil amphibian will be greatly affected by the auto-pilot and control surface characteristics.

9-3.2 FUTURE HYDROFOIL AMPHIBIANS

Projections of the characteristics and performance of future hydrofoil amphibians

TABLE 9-10 LVH-X1 VEHICLE CHARACTERISTICS

Manufacturer	Lycoming Div. of AVCO Corporation
Width, ft-in.	10-10
Length, ft-in.	36-11
Height, ft-in.	11-3
Min Ground Clearance, in.	18
Draft (foilborne), ft	2 fwd; 3½ aft
Wheelbase, ft-in.	19-4
Track Width, ft-in.	8-2
Angle of Approach, deg	33
Angle of Departure, deg	30
Break Angle, deg	15
Land Speed, mph	40
Water Speed (Design), knots (see par. 9-3.1.3 for actual)	Hullborne - 12; flying - 30; takeoff - 12.5
Land Range (Design), mi	295 at 25 mph
Water Range (Design), N.mi	Boating - 53 at 11.5 knots; flying - 195 at 35 knots
Turn Radius (Land), ft	44, 2-wheel steer
Turn Radius (Water) (Design), ft	50 boating; 370 flying
Cargo Capacity, lb	10,000
Cargo Space (L×W), ft-in.	13-7 × 10-2
Total Weight (Gross), lb	43,055
Grade Ability (Design), %	60
Side Slope (Design), %	30
Surf Ability, ft	10
Engine Data	Lycoming TF-1460 B1A Gas turb.; compr. ratio = 6.5; *BHP* at 80°F = 1000 (cont'd), 1,225 (intermittent)
Fuel Type	JP-5 or Marine Diesel
Fuel Capacity, gal	500
Marine Propulsor	Flying: (1) propeller on rear strut, D = 29 in., P = 25 in. Hullborne: steerable, retractable propeller in Kort nozzle; D = 22 in., P = 17 in.
Suspension	Oleo strut, retractable on land, with tire inflation system, 4 × 4 or 2 × 4
Brake System	Hydraulic discs
Tires	18 × 25 in., 12-ply (nylon)
Land Steering	Hydraulic selective, 2-wheel steering
Marine Steering	Foil flaps and directed propeller thrust
Electrical System	24-V DC
Hydraulic System	3,000 psi, 20 gpm; 500 psi, 10 gpm
Crew	2
Hull Material	Aluminum
Foil Type	Submerged
Foil Retraction	Hydraulic-actuated, rotary struts

TABLE 9-11. LVH-X2 VEHICLE CHARACTERISTICS

Manufacturer	FMC (San Jose)
Width, ft-in.	10-6
Length, ft - in.	38-0
Height, ft-in.	11-3
Min Ground Clearance, in.	20-3/8
Wheelbase, ft-in.	28-2-3/8
Track Width, ft-in.	8-6
Angle of Approach, deg	50
Angle of Departure, deg	28
Break Angle, deg	15
Land Speed, mph	41
Water Speed (Design), knots (See par. 9-3.1.4.1)	Hullborne - 11; flying - 35
Land Range (Design), mi	250 at 25 mph
Water Range (Design), N.mi	(Flying) 210 at 35 knots
Turn Radius (Land)(Design), ft	35, 4-wheel steer
Turn Radius (Water)	Hullborne (see par. 9-3.1.4.3)
Cargo Capacity, lb	10,000
Cargo Space (LXW), ft-in.	13-4 X 10-1-5/8
Total Weight (Gross), lb	39,000
Grade Ability (Design), %	60
Side Slope (Design), %	30
Surf Ability, ft	10
Engine Data	Solar TS-1000 gas turb.; 1,040 *BHP* (cont'd) at 22,300 rpm and 80°F
Fuel Type	JP-5
Fuel Capacity, gal	430
Marine Propulsor	(1) Propeller; D = 26 in., P = 31.2 in., MWR = 0.62; 3-bladed
Suspension	Retractable airspring; hydraulic shock absorbers; 2- or 4-wheel drive
Brake System	Hydraulic-actuated spot and disc type
Tires	18 X 25 in.; 2-ply
Land Steering	Hydraulic selective, 2-, 4-wheel, or oblique
Marine Steering	Rudder on aft strut plus fwd and aft foil flaps, rudder and wheels in surf mode
Electrical System	24-V DC, single-wire system
Hydraulic System	(1) 3,000 psi at 2,400 rpm; (2) 15,000 psi at 2,000 rpm
Crew	2
Hull Material	Aluminum
Foil Type	Surface-piercing
Foil Retraction	Hydraulic motor-driven ball screws (vertical retraction)

TABLE 9-12. LVH-X2 POWER LIMITED SPEEDS

Sea State	0	1	2	3
Speed, mph	42	38	36	33

have been made and are presented in Refs. 4 and 7. The characteristics and performance of a hydrofoil amphibian will depend largely on the requirements, restrictions, and type of foil system used. As a result, it is not possible to present as generalized information on performance in calm and rough water as for a planing hull. Fig. 9-26 illustrates this problem. In Fig. 9-26 the sustained speeds of a number of hydrofoil boats are presented as a function of nondimensional wave height. In calm water, the top speed is a function of installed power, foil configuration, take-off speed, and cavitation limits—and thus varies widely for the craft shown. At nondimensional wave heights ($H_{1/3}/\nabla^{1/3}$) above 0.3, the speeds of all craft shown are affected by motions. The PT 20 and PT 50 are craft with surface-piercing foils and no auto-pilots, and thus are most affected by waves. The LVH-X2 and the DENISON have surface-piercing foils and auto-pilots, and thus are less affected. The DOLPHIN with submerged foils and an auto-pilot has the highest sustained speeds in high sea states. Thus the foil system for a hydrofoil amphibian should be chosen only after careful consideration of all the mission requirements. If it is necessary to make a performance estimate for a given foil configuration, use should be made of references such as Ref. 19.

Ref. 7 presents a parametric study of the characteristics of hydrofoil high speed amphibians. The craft considered were based on the use of surface-piercing foil systems of two configurations. The basic vehicle and foil configurations are presented in Figs. 9-27

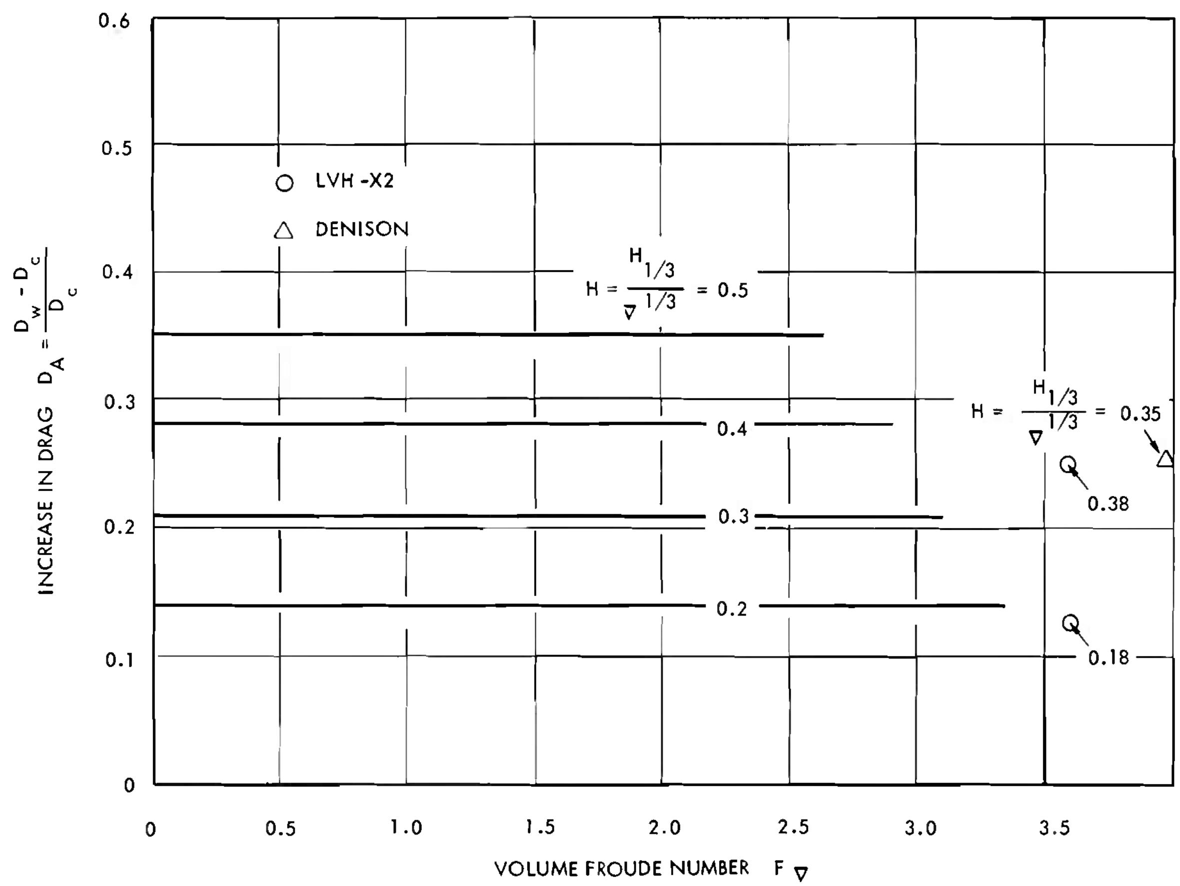


Figure 9-24. Hydrofoil Added Drag in Waves

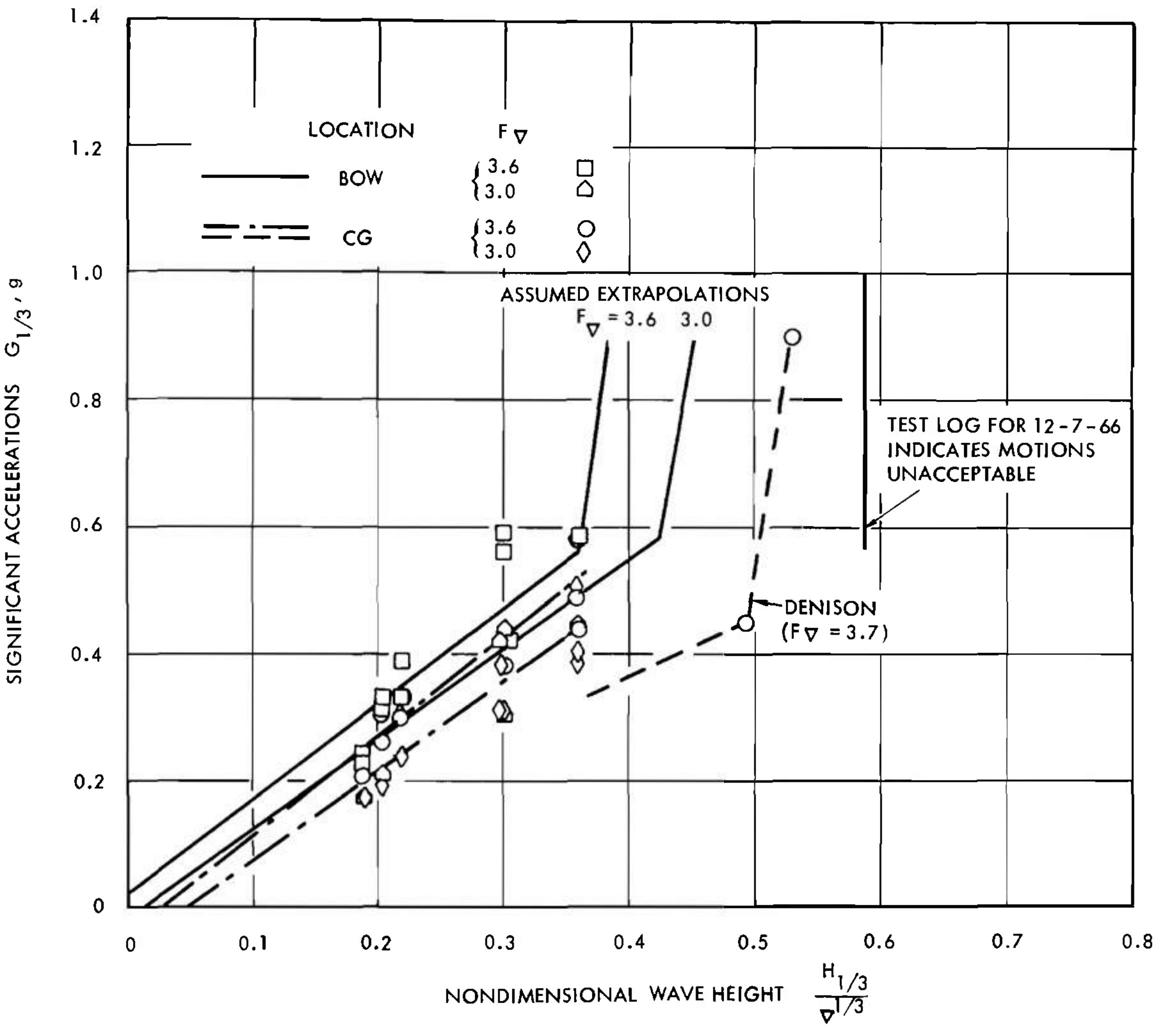


Figure 9-25. LVH-X2 Vertical Accelerations in Head Seas

through 9-30. Configuration 1 is intended to provide minimum drag at some sacrifice in motion in waves and maneuverability. Configuration 2 is intended to provide minimum motions in waves and maximum maneuverability at some sacrifice in drag. It is probable that a future hydrofoil amphibian design would be between the two extremes represented by Configurations 1 and 2. Thus the performance and characteristics of these two configurations define the range to be expected of future hydrofoil amphibians.

A parametric study was conducted to determine the effects of design payload and range on the characteristics of the two configurations. The results are presented in Fig. 9-31. The power estimates were based on a power limited speed of 40 knots in a Sea State 2. The craft were assumed to be propelled with a gas turbine engine with a specific weight of 2 lb/hp and a fuel rate of 0.75 lb/hp-hr. As indicated above, the characteristics presented in Fig. 9-31 for the two configurations span the range that may be expected with current (Summer 1968), state-of-the-art, subcavitating, surface-piercing foils.

9-4 SURFACE EFFECT AMPHIBIANS

In addition to the vehicle types already considered, it is possible that the surface effect concept may be suitable for use in a high water speed wheeled amphibian. In a surface effect craft the lift is provided by a region of

TABLE 9-13 LVH-X2 HIGH-SPEED TURNING DIAMETER

Direction	*Diameter, ft*	*Time, sec*	*Average Speed, mph*	*Turn Diameter/Vehicle Length*
Starboard	920	55	35	24.2
Port	580	40	31	15.3
Starboard	700	42	35	18.4
Port	590	39	32	15.5

TABLE 9-14 LVH-X2 SURF MODE TURNING DIAMETERS[1]

	Speed, mph	*Diameter, ft*	*Trim,[4] deg*	*Heel, deg*	*Turn Diameter/Vehicle Length*
Port	3.4[2]	126	2.0	3 stbd	3.31
Starboard	3.4	96	2.0	2 port	2.52
Port	7.0	134	3.0	7 stbd	3.52
Starboard	7.0	106	3.0	7 port	2.79
Port	9.3	153	4.0	8 stbd	4.03
Starboard	9.3	107	4.0	8 port	2.82
Port	10.8[3]	158	6.5	9 stbd	4.16
Starboard	10.8	111	6.5	9 port	2.92

Notes: (1) Four-wheel steer; GVW = 39,500 lb, two (2) personnel on board, KG = 10.3 ft load VCG approximately 20 in. above deck
(2) Minimum practical operating speed recommended by vehicle manufacturer
(3) Maximum speed obtainable in the surfing mode
(4) Trim by stern (static trim = 1.0 deg)

pressurized air contained under the craft. As a result, at high speeds the wavemaking drag and frictional drag are greatly reduced from that of a conventional form. However, as with the planing hull and hydrofoil, wavemaking drag may be large enough at intermediate speeds to cause a hump in the drag curve. The extent of this hump drag region will depend on the vehicle configuration and the air cushion pressure.

There are two basic types of surface effect craft that have been studied and developed. One of these is the rigid sidewall-type and the other the flexible skirt-type. In the sidewall-type, the air cushion is contained between two rigid sidewalls that are in contact with the water surface. The fore and aft ends of the air cushion may be sealed with flexible or rigid seals. Since the sidewalls remain in contact with the surface, this type of craft is not amphibious unless equipped with wheels. In the flexible skirt-type, the air cushion is contained by an inflated skirt around the periphery of the craft. These skirts are currently made from a neoprene-coated nylon or Dacron fabric. In general, the flexible skirts do not contact the surface but have a small clearance. As a result, these craft are amphibious if propelled by air screws or by wheels.

9-4.1 RIGID SIDEWALL SURFACE EFFECT AMPHIBIANS

Two concepts have been proposed for rigid sidewall surface effect craft that could be applied to wheeled amphibians. The first is the hydrokeel concept in which the air cushion is

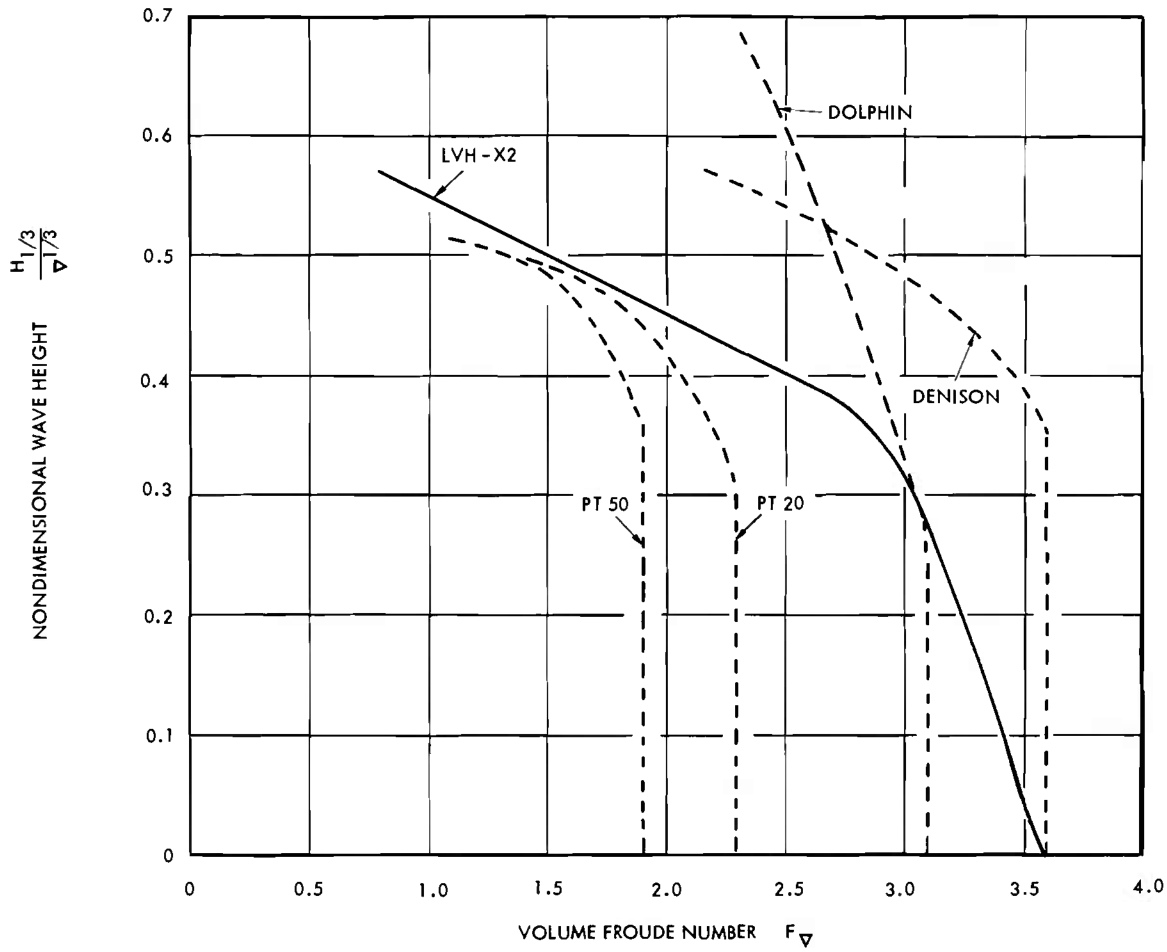


Figure 9-26. Hydrofoil Motion Limited Speeds

contained between two shallow and thin sidewalls, a flap forward and a conventional planing surface aft. The other is the Captured Air Bubble (CAB) concept in which the air cushion is contained between two deep and thick sidewalls, and flexible flaps at the forward and aft ends.

9-4.1.1 Hydrokeel Amphibians

Considerable effort has been devoted by the U. S. Marine Corps to the study of the application of the hydrokeel concept—a type of sidewall air cushion vehicle—to a wheeled amphibian. As part of this effort a hydrodynamic test craft called the ARC(K)-X1 was built and tested. This craft was intended only to study waterborne performance and thus was not equipped with wheels. To date (1969) no actual hydrokeel amphibian has been built.

The characteristics of the ARC(K)-X1 are presented in Fig. 9-32. The results of tests on the ARC(K)-X1 are summarized in Ref. 4. The measured drag of the ARC(K)-X1 in calm and rough water is presented in Fig. 9-33. The percentage increase in drag in rough water is presented in Fig. 9-34. A comparison of these data with similar data previously presented for the planing hull and hydrofoil amphibians indicates that the hydrokeel has lower drag in calm water. However, the drag of a hydrokeel increases more rapidly as wave height increases so that in a State 3 Sea a 40,000 lb hydrokeel has higher drag than the planing hull and hydrofoil.

Measurements were made of acceleration of the ARC(K)-X1 in head seas. The resulting data

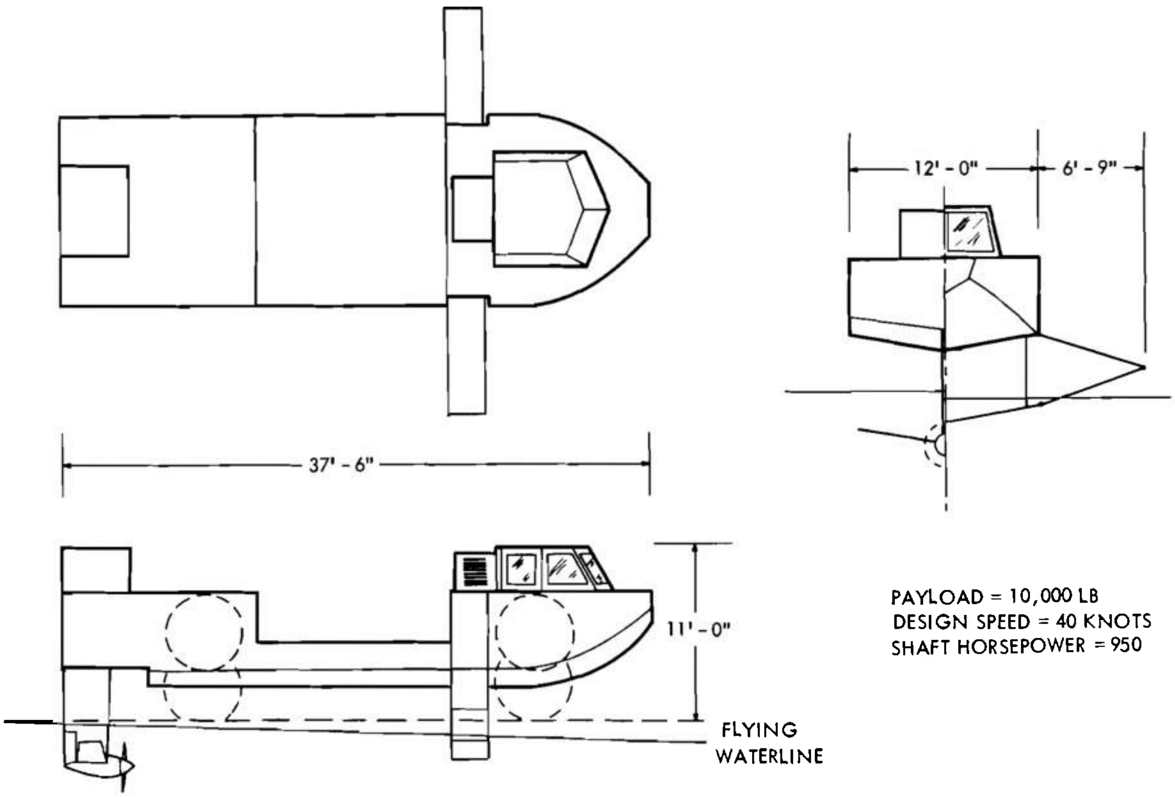


Figure 9-27. High-speed Amphibian—Hydrofoil Type, Configuration 1

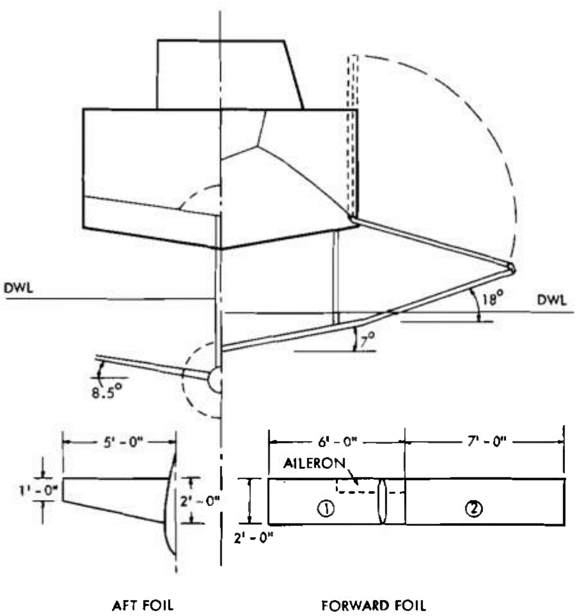


Figure 9-28. High-speed Amphibian—Hydrofoil Type, Configuration 1, Principal Foil Dimensions

are presented in Fig. 9-35. Data are also included in this figure on the acceleration of the LCVP(K), another hydrokeel of about the same size and characteristics as the ARC(K)-X1. Fig. 9-35 indicates that the accelerations of a hydrokeel amphibian will be similar to those of a planing hull amphibian and considerably greater than those of a hydrofoil amphibian.

Studies were also conducted to determine the stability and control characteristics of the ARC(K)-X1. Based on those studies it was concluded that the stability and control characteristics of a hydrokeel amphibian would be satisfactory. However, it was determined that a hydrokeel amphibian would be much more sensitive to the placement of cargo center of gravity than the planing hull or hydrofoil. Also it was felt that for stability in high speed turns a hydrokeel amphibian should be equipped with sponsons similar to those on the ARC(K)-X1.

A parametric study was conducted in Ref. 7 to determine the expected characteristics of a hydrokeel amphibian. The base vehicle developed in the study is shown in Fig. 9-36.

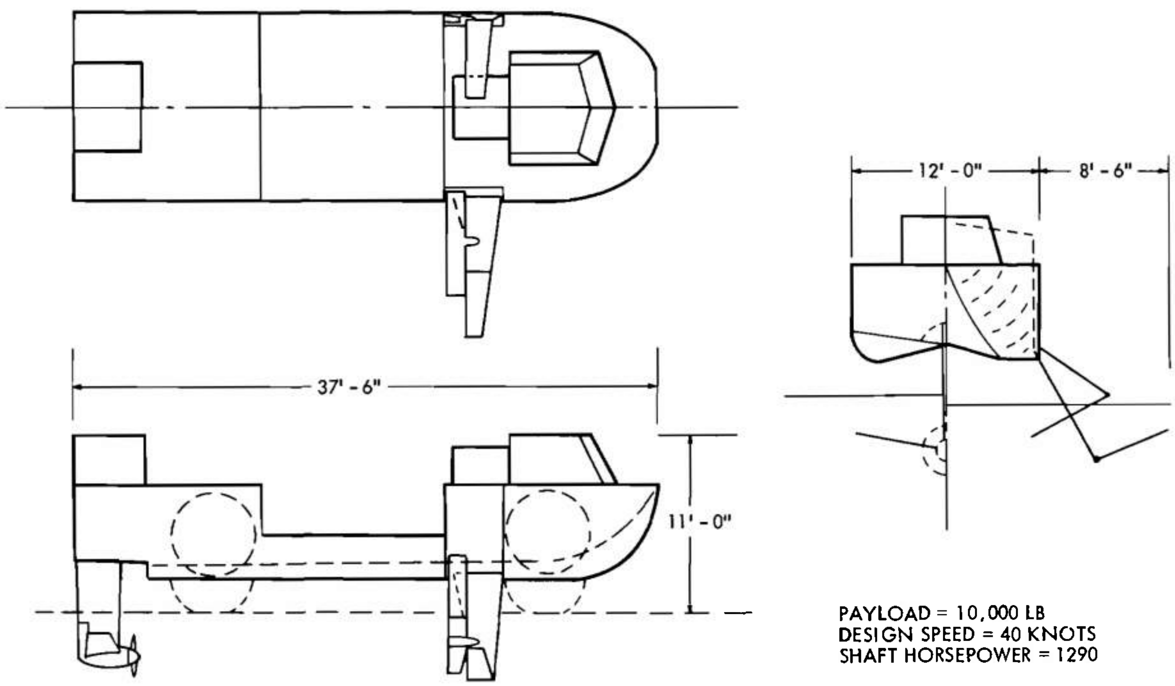


Figure 9-29. High-speed Amphibian—Hydrofoil Type, Configuration 2

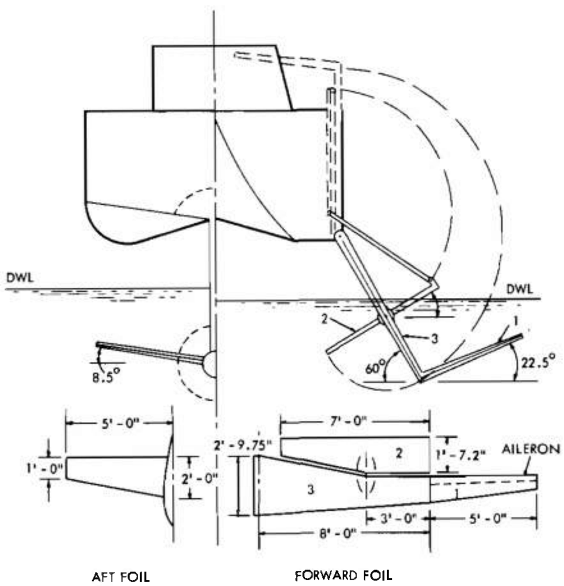


Figure 9-30. High-speed Amphibian—Hydrofoil Type, Configuration 2, Principal Foil Dimensions

This craft is intended to reach its power limited design speed in a Sea State 2. The effects of range and payload on the characteristics of a hydrokeel amphibian are presented in Fig. 9-37. These characteristics are based on the use of a gas turbine propulsion system with a specific weight of 2 lb/hp and a fuel rate of 0.75 lb/hp-hr. Since no actual hydrokeel amphibian has been built, the estimated characteristics shown in Fig. 9-37 are less certain than those given above for future planing hull and hydrofoil amphibians.

9-4.1.2 Captured Air Bubble (CAB) Amphibians

Very large surface effect craft of the CAB-type have been studied in some detail. However, no test craft have been built of a size and displacement suitable for a wheeled amphibian vehicle. There are theoretical procedures reported in Ref. 20 that can be used to make an estimate of the performance of a CAB amphibian. Such estimates were made and a parametric study of CAB amphibian characteristics are reported in Ref. 7. The base

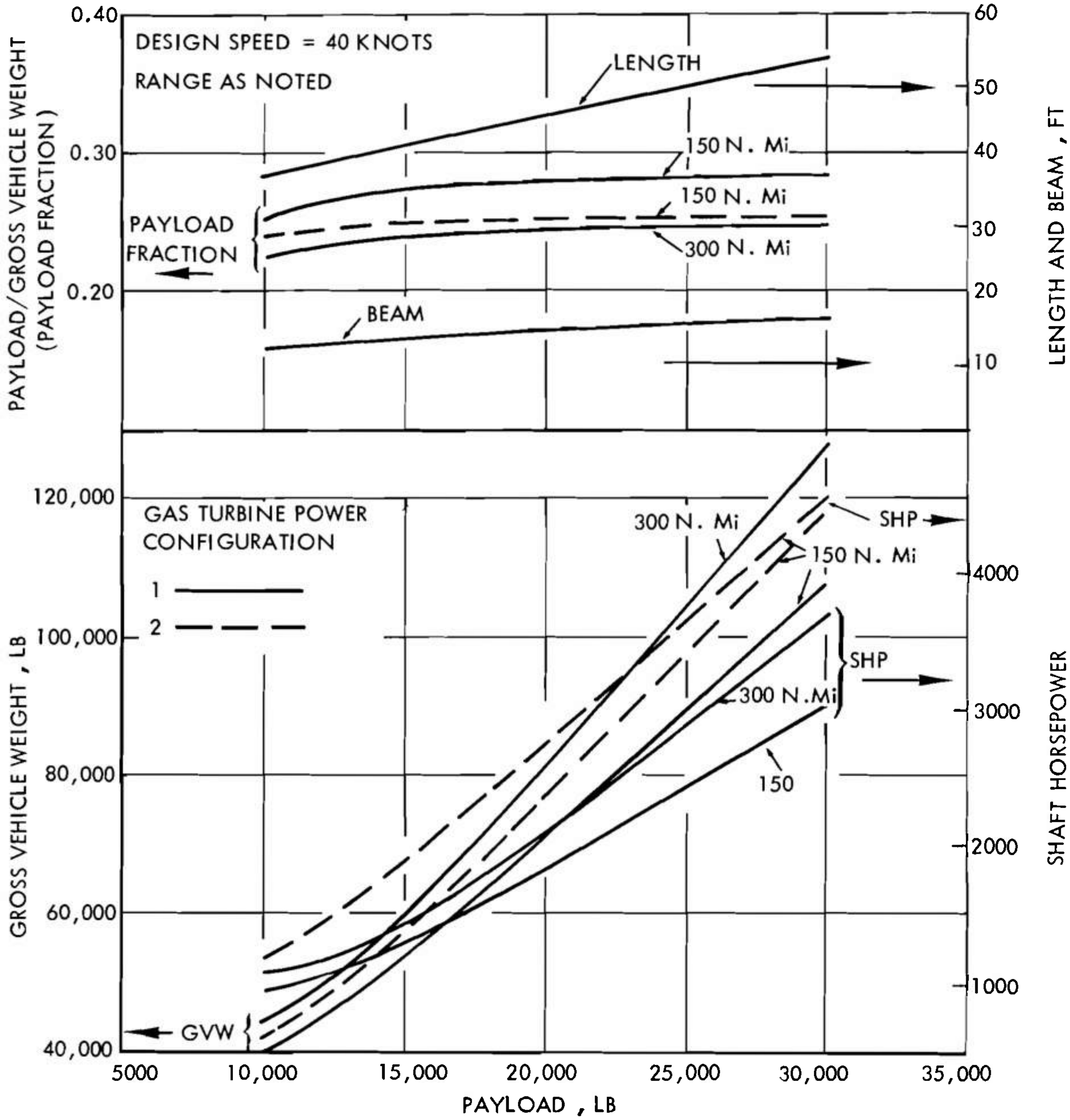


Figure 9-31. High-speed Amphibian—Hydrofoil Type, Effect of Design Payload

DIMENSIONS

SHIP

KEEL LENGTH L_K	32.30 FT
LOA	36.00 FT
BEAM I	10.35 FT
BEAM II	11.91 FT
L_K/B_I	3.12 FT
L_K/B_{II}	2.71 FT

PROPELLERS

NUMBER (2)	4177-4178
DIAMETER	26.00 IN.
PITCH	32.50 IN.
NO. OF BLADES	3
APPROXIMATE BAR	0.52
P/D	1.25
DIRECTION OF ROTATION	OUTWARD
STYLE	MICHIGAN WHEEL CO. DYNA-JET

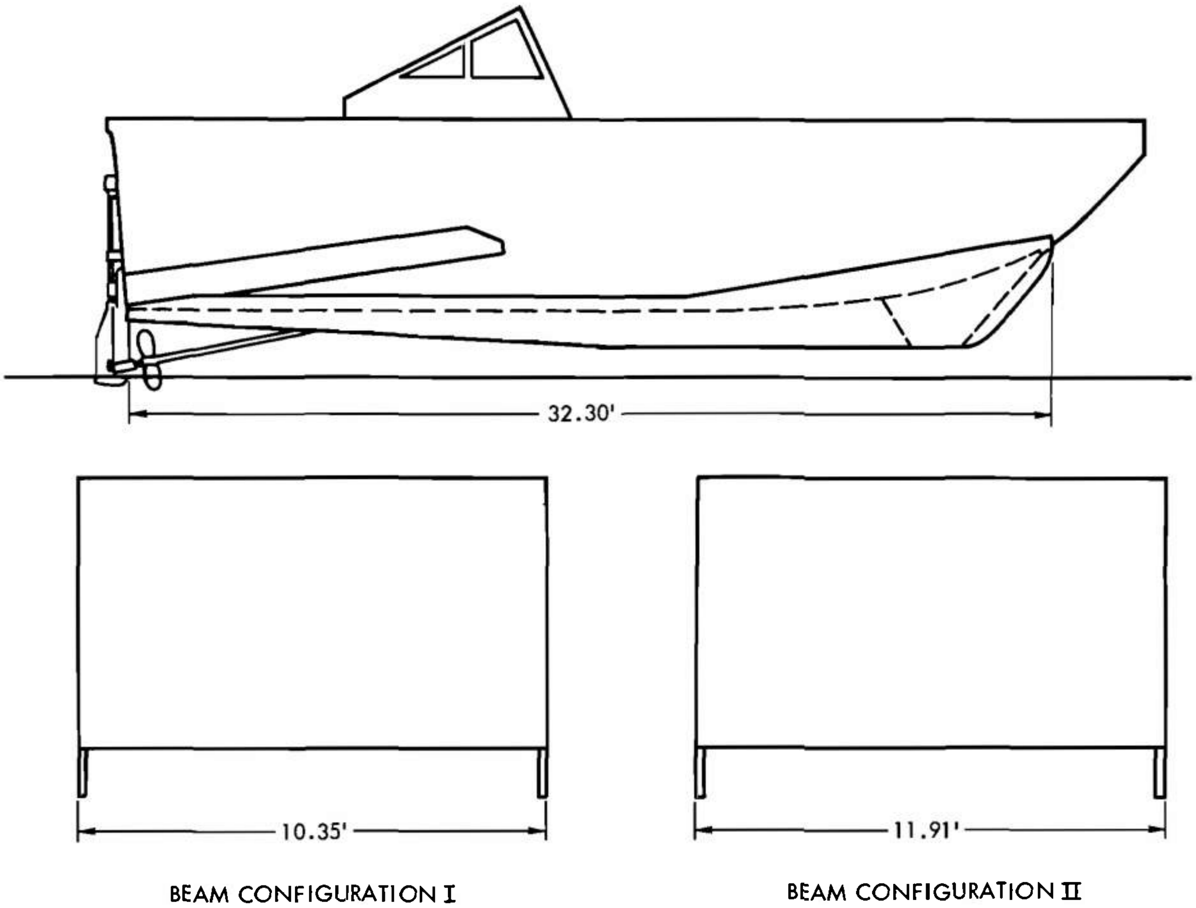


Figure 9-32. ARC(K)-X1 Dimensions

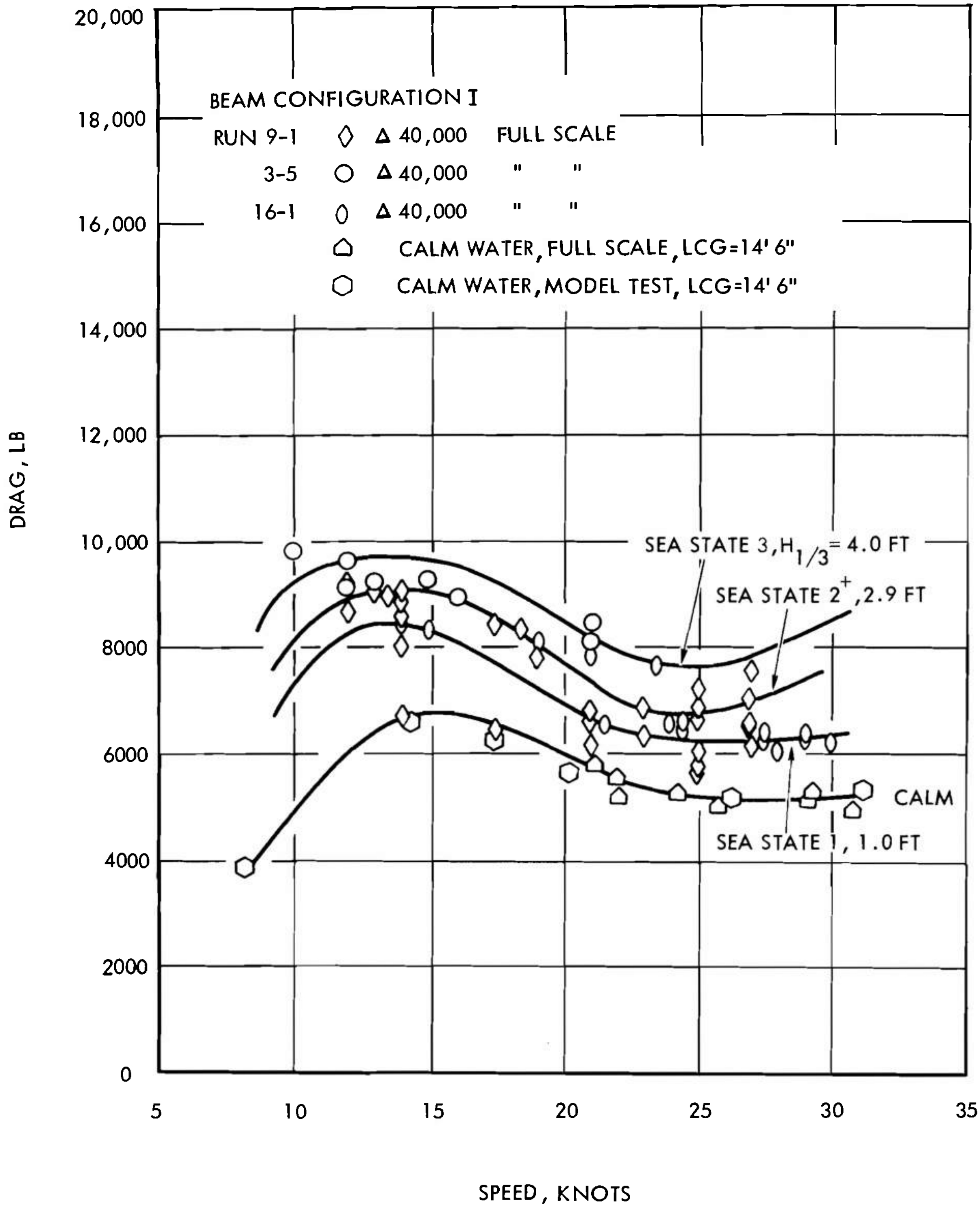


Figure 9-33. ARC(K) Rough Water Data

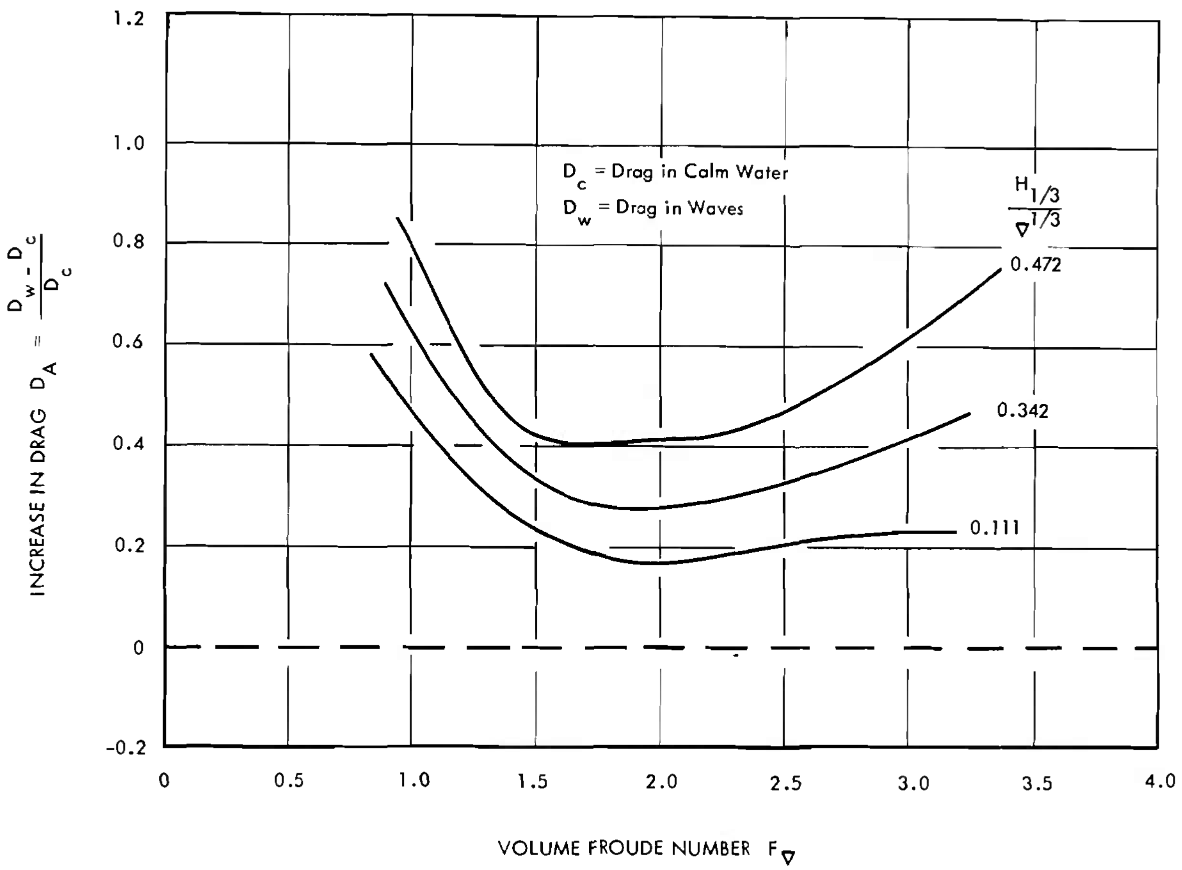


Figure 9-34. ARC(K) Added Drag in Waves

vehicle developed in this study is shown in Fig. 9-38. The effect of range and payload on craft characteristics is presented in Fig. 9-39. The most notable characteristic of a CAB amphibian is the wide beam required for stability in the "on bubble" condition. This has the effect of greatly increasing the empty weight of the craft. It should be noted that the characteristics given in Fig. 9-39 are very approximate because of the lack of experimental and prototype data.

9-4.2 FLEXIBLE SKIRT SURFACE EFFECT AMPHIBIANS

As previously indicated, the flexible skirt surface effect craft has an amphibious capability without wheels if it is propelled by an air screw. A number of vehicles of this type have been proposed for the amphibious support mission (Ref. 21). The advantages of this type of craft include a very high water speed (70 knots or more), excellent mobility over open flat ground regardless of the soil conditions, and the ability to land on a beach through very high surf (i.e., they can outrun the surf, see par. 7-4). The disadvantages of this type of craft include its large dimensions relative to the payload carried, its poor maneuverability in restricted places, its limited ability to climb grades, and its limited ability to exit from a beach through high surf. To take advantage of the characteristics of this type of vehicle, changes in the operational techniques now used with wheeled amphibians would be required.

In order to avoid some of the disadvantages of air propellers on flexible skirt surface effect vehicles, studies have been made of the possibilities of using wheels for partial support and propulsion on land (Ref. 22). No vehicle of this type has been built that could provide data

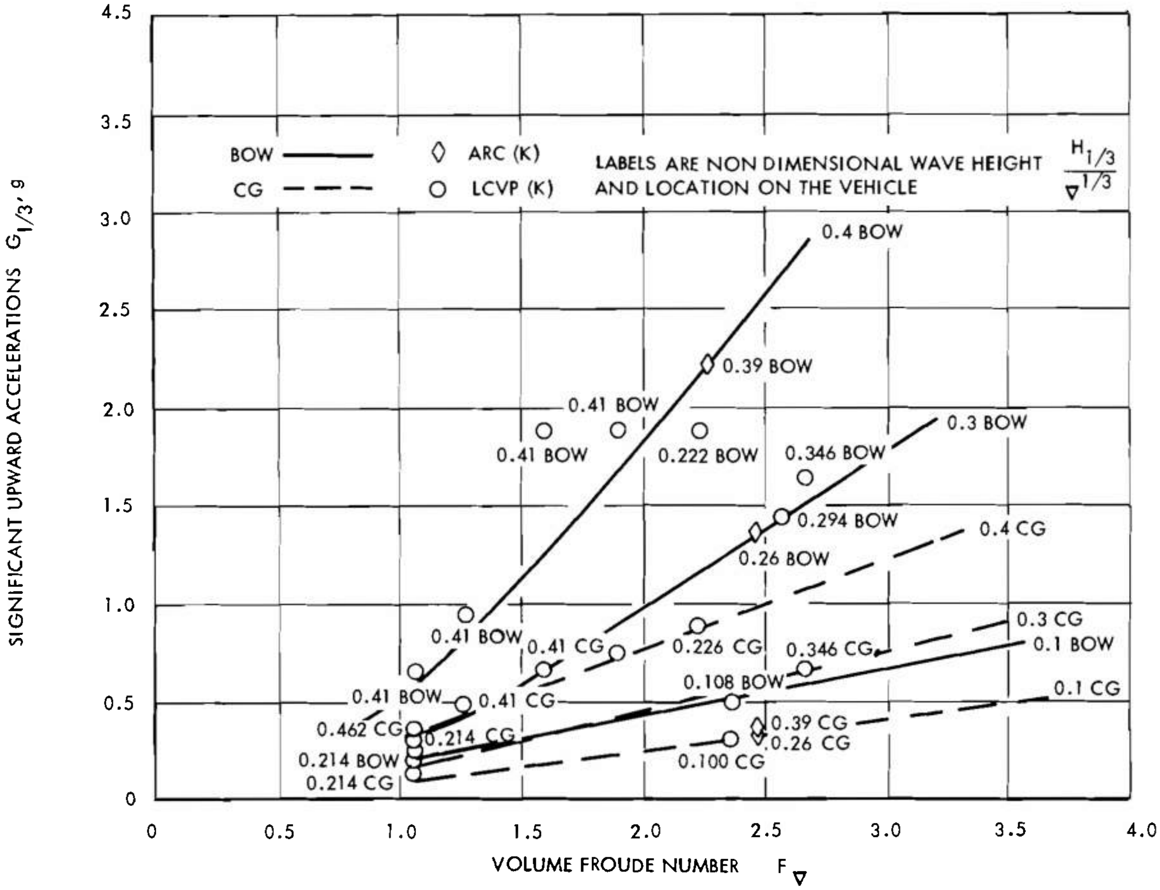


Figure 9-35. Hydrokeel Accelerations in Head Seas

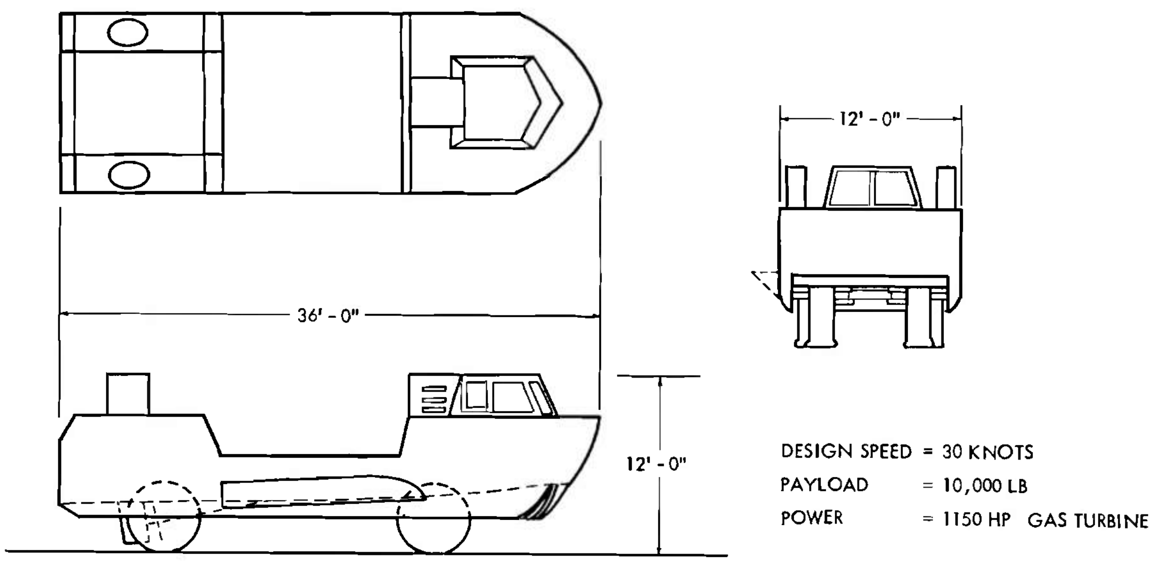


Figure 9-36. High-speed Amphibian—Hydrokeel Type, Base Vehicle Configuration

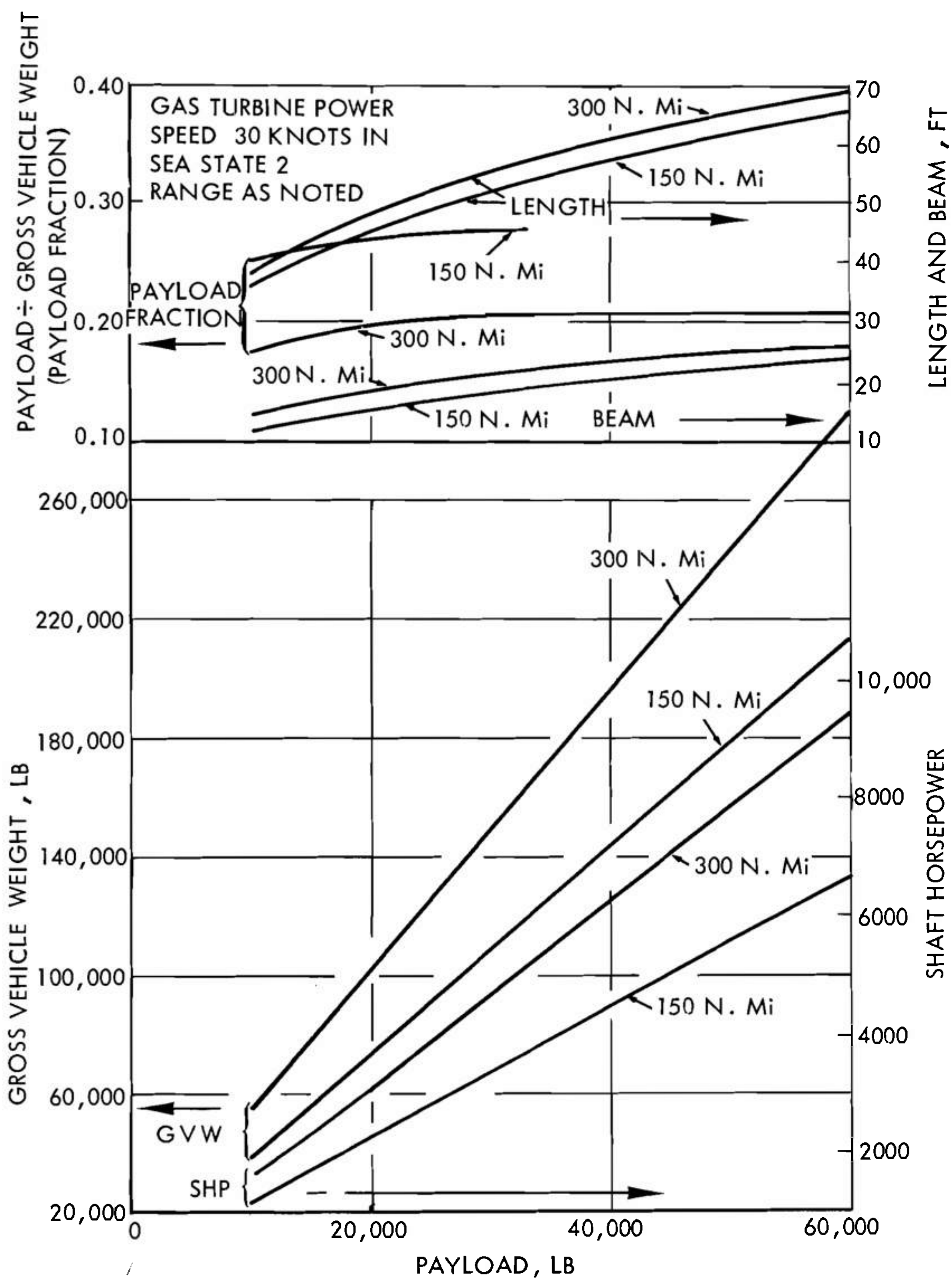


Figure 9-37. High-speed Amphibian—Hydrokeel Type, Effect of Payload

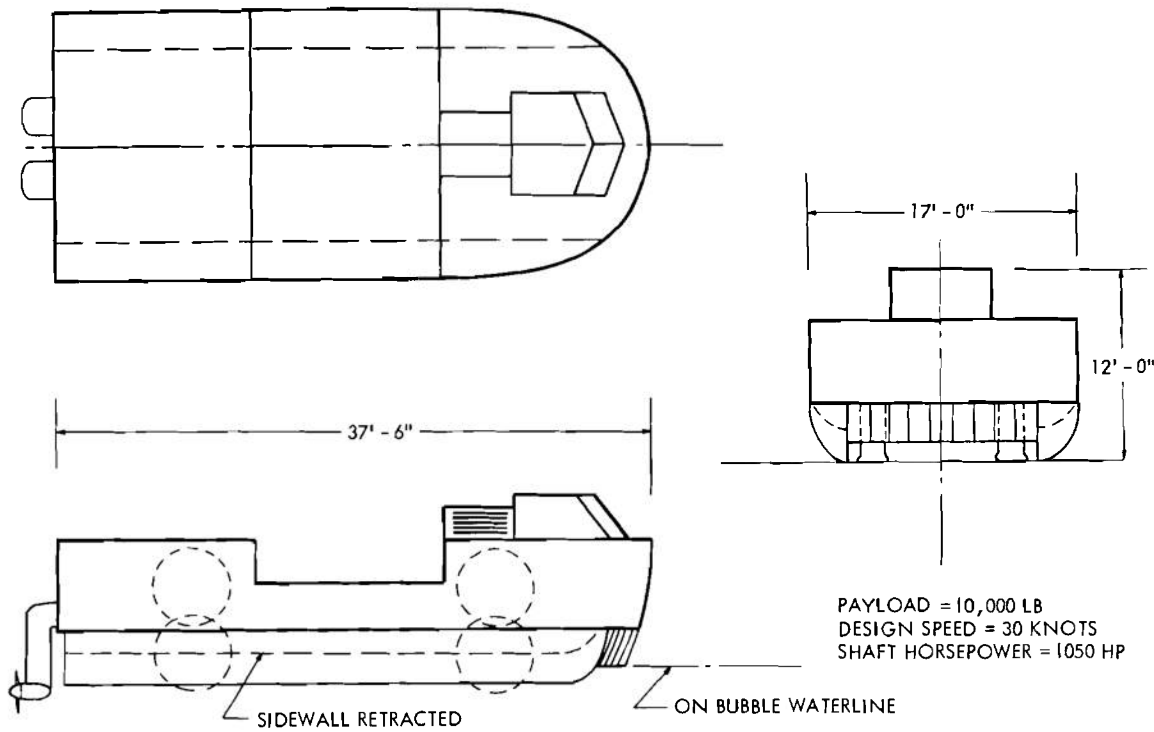


Figure 9-38. High-speed Amphibian—CAB Type

for application to a high-speed wheeled amphibian. However, a prototype air cushion truck has been built in France in which the wheels supply propulsion and carry 5 to 20 percent of the weight on land (Ref. 23). This vehicle has about the same dimensions as a LARC V and carries about 6,000 lb payload. It has the ability to "swim" in protected waters at speeds up to 4.5 mph. Future developments with this type of vehicle should be reviewed by the designer since they may have direct application to high-speed wheeled amphibians.

9-5 DISPLACEMENT FORM HIGH-SPEED AMPHIBIANS

All of the possible high-speed amphibian types considered so far depend on dynamic forces for lift at high speed and are characterized by high drag at intermediate speeds. As a result, there is a speed gap between conventional amphibians with a top speed of 10 to 12 mph and high-speed amphibians with minimum practical speeds of 18 to 20 mph. Some missions may require an amphibian that can operate economically in this intermediate speed range.

In a displacement type vehicle, high water speeds can be achieved at moderate power levels if low displacement length ratio hull forms are used. Because of the large size required for such a concept, a single hull would not be practical for an amphibian. However, a catamaran hull form could be developed, with each hull having an acceptable displacement-length ratio, and still remain within an acceptable size. Such a configuration is illustrated in Fig. 9-40. In this concept the overall length, for land operations, can be reduced by folding the bow and stern sections of the hulls back on the mid-body.

9-5.1 CHARACTERISTICS

Since the catamaran is a displacement type, it will not have a high hump drag at moderate speeds. As a result, moderate speeds should be possible with low installed powers. The configuration shown in Fig. 9-40 is estimated to have a top speed in calm water of 22.5 knots with 600 hp delivered to the propeller shaft. The drag estimate was based on the data presented for the Series 64 high-speed displacement hull

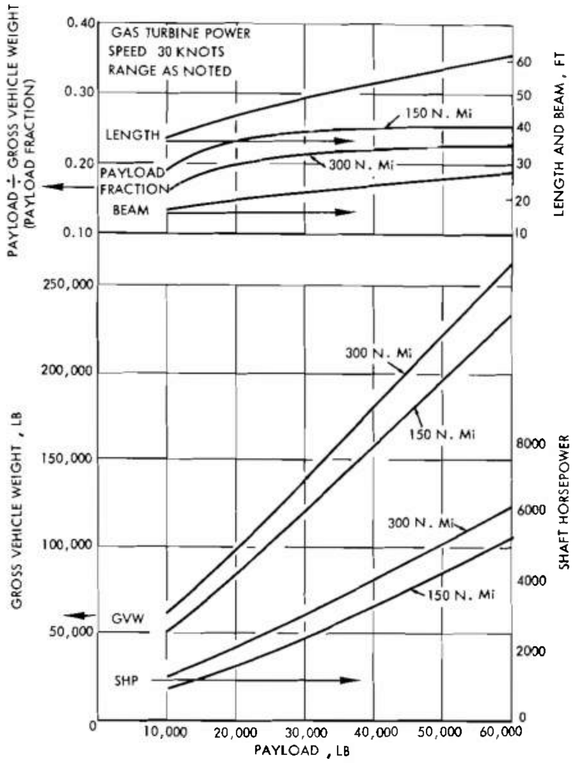


Figure 9-39. High-speed Amphibian — CAB Type, Effect of Payload

forms. An additional advantage of this type of hull form is that it should have good seakeeping characteristics because of the long slender hulls.

The catamaran amphibian shown in Fig. 9-40 has a design payload of 10,000 lb. The characteristics of this concept are presented in Table 9-15. As already noted, the bows and sterns fold back for land operations. The wheels would retract into the hulls for high speed water operation. This is a feature the catamaran has in common with all of the high speed amphibian types. Waterborne, propulsion is provided by a single propeller out-drive unit, rotatable to provide steering control. At low water speeds the wheels will also provide some steering control if in the extended position.

The catamaran configuration is not suitable for surf operations because of the small buoyancy of the fine bows. The forward end will tend to dig into the back of a breaker, causing the vehicle to broach. There is also a danger of the fine bows or sterns hitting the bottom as the vehicle pitches in a breaker. To avoid these problems, the catamaran amphibian must cross the surf with the bows and sterns folded back. In this configuration the catamaran amphibian should act like the other high speed amphibian types in the surf. Also, the roll angle

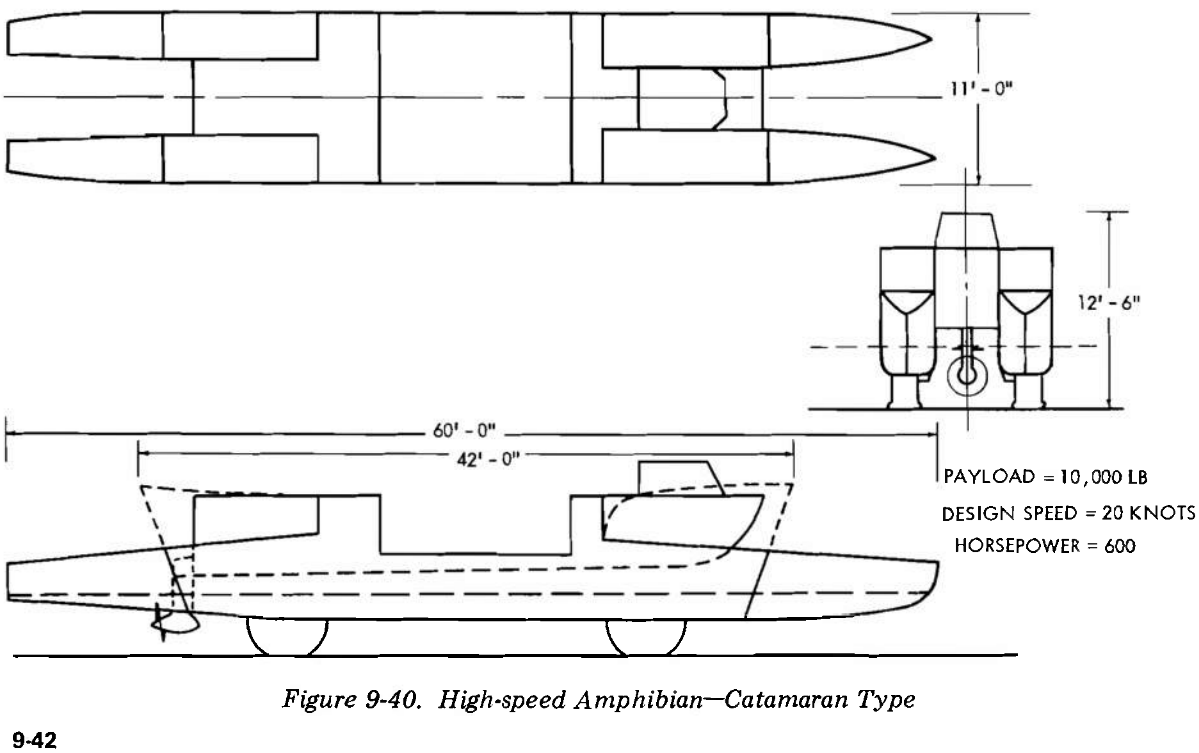


Figure 9-40. High-speed Amphibian—Catamaran Type

to capsize (range of stability) would be increased somewhat since the folded bows and sterns have the effect of increasing the freeboard.

The advantages of moderately high speeds with low power, and good seakeeping out of the surf zone, make a catamaran type amphibian appear attractive. There are, however, a number of disadvantages with a catamaran form. The major disadvantage involves the arrangement of internal components. There is very little internal volume in a catamaran suitable for the machinery and drive train components. A detailed study would be required to determine if a satisfactory arrangement could be developed.

TABLE 9-15 HIGH-SPEED SUPPORT AMPHIBIAN–CATAMARAN TYPE

Payload - 10,000 lb

Power Limited Speed - Calm Water, 22.5 knots
Sea State 3, 20 knots

Length, Overall
Hull, Unfolded - 60 ft
Hull, Folded - 42 ft
Breadth - 11 ft
Height - Wheel Down, 12.5 ft
Wheels Up, 10.5 ft

Power - 600 horsepower gas turbine

Range - 150 N.mi

Water Propulsion - single propeller out-drive

Gross Vehicle Weight - 40,000 lb

Weight Summary

Item	*Weight, lb*
Hull Structure	10,000
Folding Bows and Sterns	3,000
Machinery Weight	1,200
Main Drive Train	2,800
Marine Drive	600
Land Gear	6,000
Misc. Systems	2,500
Empty Weight	26,100
Crew	500
Fuel	3,400
Payload	10,000
Gross Weight	40,000

The folded bows will cause a visibility problem from the cab when on land unless the vehicle height is increased above that shown in Fig. 9-40. The overall length of the catamaran amphibian may be a disadvantage for land operations. The configuration shown in Fig. 9-40 is 2 to 6 ft longer than existing high-speed amphibians. The length could be reduced by reducing the intersection angle between the mid-body and the bows and sterns. This, however, would reduce the already small approach and departure angles.

9-5.2 PERFORMANCE

The major unknown from a hydrodynamic standpoint is the problem of interference drag between the two hulls. There are no experimental data on this subject which apply to configurations suitable for a catamaran amphibian. The performance estimates have been made assuming that interference drag will be small. If this is not the case, many of the justifications for a catamaran configuration are lost. The best way of determining the importance of interference drag would be through a small model test program. This is beyond the scope of this handbook and it was decided not to continue with a parametric study of a catamaran amphibian at this time. However, it is felt that the potential of a catamaran amphibian is great enough to justify a small model test program at some future date.

9-6 COMPARISON OF VEHICLE TYPES FOR USE AS HIGH-SPEED AMPHIBIANS

The major decision that must be made in the design of a high speed amphibian is the selection of the vehicle type. This selection must depend on the mission requirements since no one type is superior in all areas. Table 9-16 has been developed to aid in this selection by providing the relative rankings of planing hull, hydrofoil, and hydrokeel types in a number of areas. The other types discussed are not included in this ranking since reliable prototype or experimental data are not available for them. A rank of 1 indicates the vehicle type that is best in the particular area. The quantitative differences between vehicle types can be obtained from the previous descriptions.

TABLE 9-16 RELATIVE RANKING OF VEHICLE TYPES FOR USE AS HIGH-SPEED AMPHIBIANS

	Item	*Planing Hull*	*Hydrofoil*	*Hydrokeel*
1.	Gross Weight	1	3	2
2.	Low Speed Stability and Seaworthiness	1	1	1
3.	Sensitivity to Variation of Cargo CG Location	1	2	3
4.	Calm Water Top Speed	3	2	1
5.	Power Limited Speed in Sea State 3	2	1	3
6.	Motion Limited Speed in Sea State 3	2	1	2
7.	Maneuverability and Control at Low Speed	1	2	1
8.	High Speed Stability in a Turn	1	2	3
9.	Compatibility with Secondary Road Nets	1	1	1
10.	Complexity	1	3	2

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GLOSSARY

active rudder. A rudder fitted with a submersible motor and propeller to provide directional thrust at low speeds.
aft. Toward, at, or near the stern.
amidships. In the vicinity of the mid-length of the vehicle, as distinguished from the ends; also 'midships.
angle of approach. Angle formed by a line connecting the tangent of the front tires and the lowest portion of the vehicle body, and the horizontal.
angle of departure. Angle formed by a line connecting the tangent of the rear tires and the lowest portion of the vehicle body, and the horizontal.
appendages. Portions of the vehicle extending beyond the hull outline; rudders, shafting, struts, wheels, gear cases, etc.
area moment of inertia. Second moment of the area; for a discrete area this is the area multiplied by the square of the distance from the centroid to the axis of reference.
aspect ratio. Ratio of the chord to span of a rudder, foil, or propeller blade.
athwartship. Across the vehicle, perpendicular to the fore and aft center plane.
ballast. Any solid or liquid weight placed in a vehicle to change the trim or to regulate the stability.
baseline. A fore and aft reference line at the lower surface of the bottom plating at the center of the vehicle. The base plane is a horizontal plane through the baseline, parallel to the waterplane.
bilge. Curved section of the hull and underbody between the vehicle bottom and vertical side; recess into which the water drains from cargo and machinery spaces.
bilge pump. A pump used to remove the water and other liquids from the bilge.
bitt, mooring. Short metal column extending from a base plate on the deck, used for securing the vehicle.
block coefficient. Ratio of the actual underwater volume to the volume of a rectangular block, which has dimensions of the length, beam, and draft.
body plan. Composite drawing outlining the intersection of the section planes or stations with the hull surface.
bollard pull. Thrust delivered by the propulsor at zero forward speed.
Bonjeans curves. A plan giving the underwater area for each station or section along the vehicle as a function of draft.
bow. The forward end of the vehicle.
break angle. The angle formed by a line from a point on the hull midway between the wheels and tangent to lower edge of the tire (the angle labeled 22° in Fig. 5-7 is an example).
bulkhead. A vertical partition which divides the vehicle into compartments. Bulkheads may be nontight, water-tight, oil-tight, or gas-tight, depending on construction.
buoyancy. Upward force exerted by the fluid on a body; proportional to the volume of water displaced by the body.
buttocks (planes). Vertical longitudinal planes, parallel to the longitudinal center plane, perpendicular to the base plane and water planes.
catamaran. Twin hulled craft which is generally symmetrical about the centerline.
centerline. The middle line extending the fore-and-aft length of the vehicle. More accurately described as a center plane.
chine. Intersection between side and bottom shell, or between two bottom shell planes.
chord. The length from leading edge to trailing edge of a foil or shape, in line with the direction of flow.
coaming. A vertical plate surrounding a hatch or cargo well, serves to restrict water from flowing into the opening and to strengthen the hatch edge.
conceptual design. Phase of design in which basic vehicle characteristics, and realistic and specific requirements are developed.
correlation allowance. Incremental increase in the resistance coefficient to account for surface roughness.
cross curves of stability. A plan giving the righting arm *(GZ)* as a function of draft, for various heel angles.
curves of form. A plan giving various hydrostatic properties of the hull as a function of draft.
damaged stability. Stability of an amphibious vehicle when in the damaged (flooded) condition.

deadrise. Rise of bottom of a midship frame from the keel to the bilge.

deck. A deck or platform of a vehicle; corresponds to the floors of a building.

delivered horsepower. Power delivered to the propulsor.

depth. Distance from hull bottom to deck.

directional stability. The property of a vehicle that indicates course-keeping ability.

displacement. The weight of water the hull displaces, which equals its total weight.

draft. The distance from the static waterline to a reference plane. The reference plane usually passes through the lowest point on the vehicle.

draw bar pull. The maximum towline force the landborne amphibian can develop.

effective horsepower (EHP). Power actually delivered by the propulsor. It is defined by (DV/C) where D = drag, V = speed, C = constant; the value of C depends on the units of D and V (550 for lb, ft, sec system).

failure. A deficiency which prevents a component or subassembly from performing its intended function within previously established limits.

factor of safety. The ratio of the design load to the expected actual load of a structural member.

fair (line). A line which is smooth with no abrupt changes in the radius of curvature.

final design. Design phase, following preliminary design, in which the final plans are developed.

floor. Vertical, transverse member supporting the bottom shell plating.

fore. A term pertaining to portions of the vehicle at or adjacent to the bow.

forward (fwd). In the direction of the bow of the vehicle.

frame. A transverse or longitudinal member of the vehicle's structure which provides support for the shell plating and acts to maintain longitudinal and transverse hull shape.

frame spacing. The longitudinal distance between transverse frames. Varies with the structure and location along the vehicle's length.

freeboard. Distance from the waterline to the weather deck.

freeing port. An opening in a bulwark or coaming to permit water to drain overboard.

frictional resistance. Resistance due to viscous forces acting on the vehicle; frictional resistance is directly proportional to the density, square of the velocity, and the surface area.

Froude number (F_N). A nondimensional ratio of the speed and length; $(V/\sqrt{gL})$. *See* **volume Froude number (F_∇)**.

fuel rate. The fuel consumption per horsepower hour.

galvanic action. The production of an electric current when two dissimilar metals are immersed in an electrolyte.

galvanic series. The order of metals according to their tendency to corrode galvanically.

geosim. A model which is geometrically similar to a parent model or prototype vehicle.

girth. Any expanded length around a curved section.

gross vehicle weight (GVW). Total weight of a vehicle which includes cargo, fuel, crew, and empty vehicle weight.

ground clearance. Height above the ground of the lowest part of the vehicle.

half breadth plan. Composite drawing outlining the intersection of the waterline planes with the hull surface.

heave. Motion of a vehicle in the vertical direction.

heel. Static inclination of a ship to one side, caused by off-center loads or external forces.

hogging condition. Temporary or permanent deformation of the hull in the longitudinal plane such that the hull is deflected convex upward between hull ends.

hull. The primary structural body of an amphibian, including the shell plating, frames, longitudinals, and bulkheads.

hull efficiency. One minus the thrust deduction divided by one minus the wake fraction, i.e., $1-t/(1-w)$.

hydrokeel. A type of sidewall air cushion vehicle.

hydrostatic pressure. The pressure of the water at a given depth.

inboard. Inside the ship; towards the centerline or away from the sides.

inclining test. An experiment performed on a floating vehicle to determine the initial stability, or GM.

intact stability. Stability of the vehicle in the intact, undamaged condition.

keel. Principal fore-and-aft member of the vehicle framing, extending along the bottom centerline.

knot. Unit of speed, equals one nautical mile per hour, = 1.689 ft/sec or 1.15 mph.

length, overall. Extreme length of vehicle from maximum extent of vehicle stem and stern.

length, waterline. Length of vehicle on the water, at a given draft, excluding overhangs.
lines drawing. A plan containing the three orthogonal views of the vehicle hull geometry.
longitudinals. Fore and aft structural members.
maneuverability. Property of a marine vehicle relating to ability to control, change, or retain direction of motion and speed.
margin. An allowance for weights and center of gravity location to allow for items which might be overlooked during design or incorporated in the vehicle after design has been completed.
mass moment of inertia. The second moment of mass; for a discrete mass, the mass moment of inertia is the mass multiplied by the square of the distance from the reference axis to the center of mass, slug-ft^2.
materiel. The equipment, apparatus, and supplies required for the operation of a military installation.
metacenter. The intersection of a line drawn vertically through the heeled center of buoyancy and the ship centerline plane. The heeled center of buoyancy is not on the centerline.
metacentric height. Distance from the metacenter to the center of gravity of the vehicle, symbolically GM.
metacentric radius. Distance from the center of buoyancy to the metacenter.
midship. Midway between the ends of the vehicle; for a vehicle with twenty-one stations (0,1,.....20), the midship is station 10.
mobility. The ease with which a vehicle can move from one point to another under its own power.
offsets. Coordinates defining the geometric hull form.
open center system. Hydraulic system equipped with a relief valve which allows the fluid from the pump to empty into the reservoir when it is no longer required to run a power take-off.
outboard. Direction away from the centerline towards the side.
peel strength. The force required to peel adhesives from a surface.
pitch. Dynamic rotation of the ship about the transverse axis in the longitudinal plane.
pitch, propeller. The theoretical distance that a point on the surface of a propeller blade will advance in one revolution.
planform. Contour or outline of a body when viewed from above.
port. The left side of a ship or vehicle, when looking from aft to forward.
power train. System which transmits power from the engine to the wheels and water propulsor.
preliminary design. Design phase in which the characteristics developed in the conceptual design are checked and modified until the requirements are satisfied.
profile plan. Composite drawing outlining the intersection of the buttock planes with the hull surface.
propulsor. The system which propels the vehicle when waterborne, e.g., screw propeller, water jets, paddles.
relative rotative efficiency. A factor which accounts for the difference in performance between a propeller operating in a uniform and nonuniform wake.
reserve buoyancy. Volume of the watertight hull above the design waterlinc.
reserve stability. Area between the righting arm and the heeling arm curves on a statical stability diagram.
residuary resistance. Sum of the eddy-making and wave-making resistance.
Reynolds number. Nondimensional parameter containing speed and length parameters. For ship: $Re = VL/\nu$ where L = ship length, V = ship speed, and ν = kinematic viscosity.
roll. Dynamic angular rotation of the ship in a transverse plane about the longitudinal axis.
rolling resistance. Resistance to land motion of a moving vehicle; i.e., with tires turning.
running gear. Wheels and other related components of the amphibian land system.
sagging condition. Temporary or permanent deformation of the hull in the longitudinal plane such that the hull is deflected convex downward between hull ends.
scantlings. The dimensions and thickness of structural members.
sea state. An arbitrary scale of sea conditions, based on significant wave height.
section planes. Vertical planes which cut the hull transversely.
sheer. Longitudinal curve in the deck height in a vertical plane.
significant wave height. The average height from crest to trough of the 1/3 highest waves.
Simpson's rule. A numerical integration procedure.
soil index. Quantitative measure of some physical property of the soil which can be empirically related to vehicle trafficability.

span. The tip to tip length of a wing or foil, perpendicular to the flow.

spray strip. A strip or wedge on a planing hull, just above the waterline, which deflects water or spray downward.

stability. Tendency of a marine vehicle to remain upright or the ability of a vehicle to return to upright when heeled by wind, wave, or other forces.

starboard. The right side of a ship or vehicle, when viewed from aft to forward.

stations. Divisions of the length of a vehicle into ten, or a multiple of ten, equal spaces; also, transverse section planes at these divisions.

stem. The bow frame forming the forward sides of the vehicle.

stern. After end of the vehicle.

sulfidation. A reaction between sulfur in fuel and salt in air intake of gas turbine, forming a corrosive slag.

surface effect craft. A vehicle which is supported on a cushion of pressurized air.

surge. Motion of a vehicle; longitudinal displacement in the horizontal direction.

sway. Motion of a vehicle; transverse displacement in the horizontal plane.

tactical diameter. The steady turn diameter of a waterborne vehicle with rudder hard over.

thrust deduction. The augment of ship resistance resulting from action of the propeller in changing the pressure field around the hull.

towing pad. An eye on the front or rear of a vehicle to permit attachment of a towing line.

tractive effort. Thrust that the tires must develop to meet land performance requirements.

transom. Aft transverse shell of vehicle.

transportability. The ease with which a vehicle can be moved from one location to another by different modes of transport when not under its own power.

transverse. Perpendicular to the fore-and-aft or longitudinal axis.

trim. Difference between forward and after drafts.

volume Froude number (F_∇). A nondimensional measure of speed for a planing hull. *See* **Froude number (F_N).**

wake fraction (w). Relationship between ship speed and propeller advance speed.

waterline (planes). Horizontal planes orthogonal to the section and buttock planes.

wheelbase. Distance between the points of contact of the front and rear axles of a vehicle.

wind drag. Resistance to the vehicle's motion caused by the wind.

yaw. Dynamic rotation of a vehicle about the vertical centerline axis.

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